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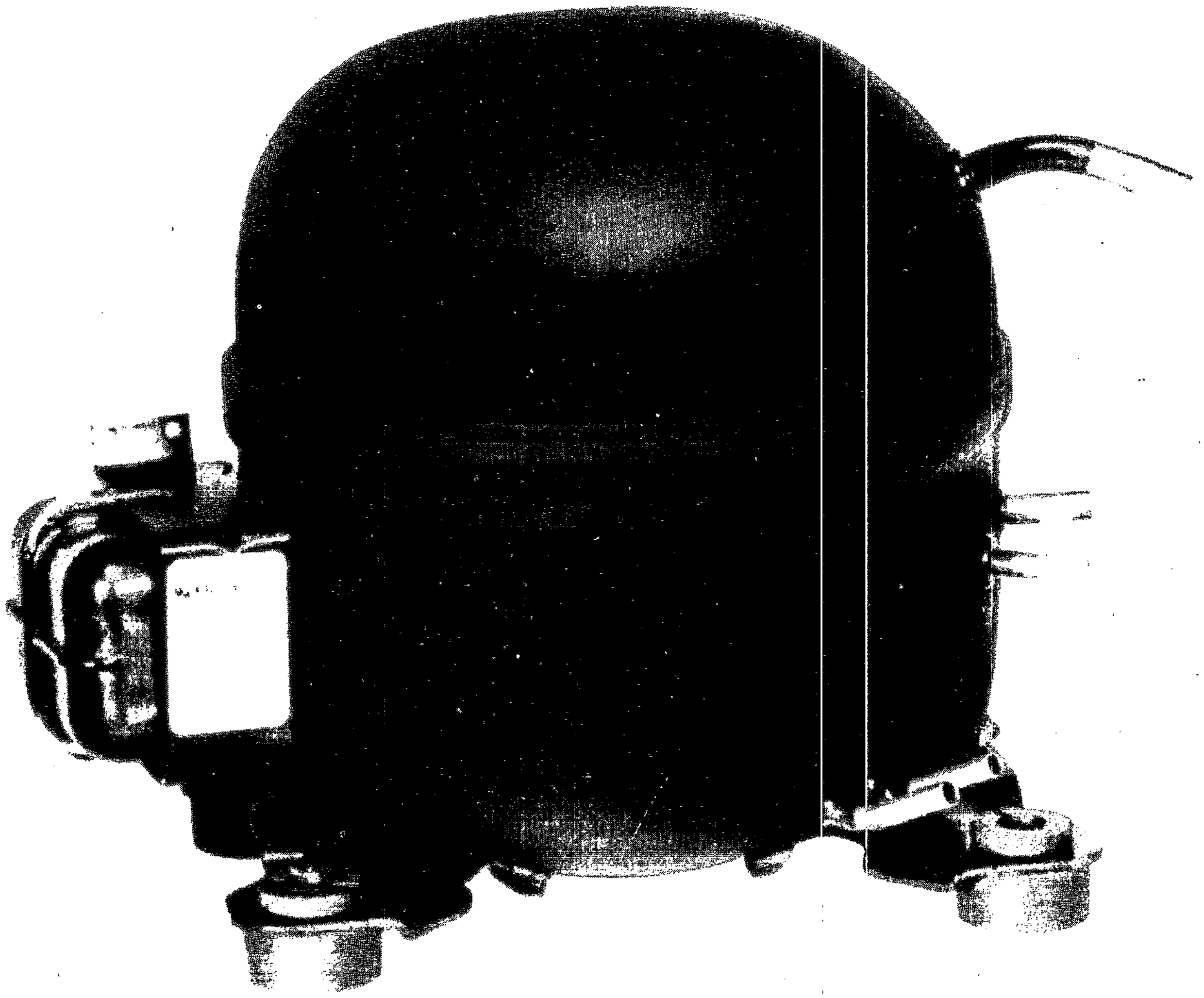
State of the Art Survey of Hermetic Compressor Technology Applicable to Domestic Refrigerator/Freezers



State Of The Art Survey Of Hermetic Compressor Technology Applicable to Domestic Refrigerator/Freezers

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Prologue

Compressor technology will play an important role in allowing potential efficiency improvements of various refrigerator/freezer design alternatives. This review of the technology of small hermetic refrigeration compressors was undertaken as part of a larger study to evaluate the options for maximizing the efficiency of domestic refrigerators. The compressor characteristics documented in the report are being used in the overall study as part of the input database for modeling and evaluating design options.

Findings

1. ***The efficiency of "large" compressors (capacity greater than 750 Btu/hr) has been improved significantly over the past decade. Further improvements are possible, at incrementally higher costs.***

- In 1980 the best "large" compressor energy efficiency ratio (EER), at standard conditions, was 4.0 Btu/Watt-hr.
- The best current production EER levels are 5.5.
- In response to 1993 refrigerator/freezer efficiency standards, compressor models with significantly upgraded efficiency are becoming available (for use with CFC-12).
- Compressors with nominal capacity above 750 Btu/hr, with EER levels up to 6.0 will be commercially available to OEM's in 1993.
- This represents approximately a 6% increase in the efficiency level of commercially available compressors in this capacity range. About one half of this increase is attributable to higher motor efficiencies, the remainder to incremental reductions in mechanical and thermal losses.
- An EER level of approximately 6.5 is technically feasible, at an incremental increase in OEM costs on the order of $15.

2. ***The value of these energy savings was calculated.***

- Assuming an electric energy cost of 8¢ per KwH, for real discount rates less than 10%, the real, present value of the saved energy is on the order of $100, over the projected 15 year life of the refrigerator, much greater than the incremental cost, of the increased efficiency compressor.

3. ***Smaller compressors will be needed in future refrigerators.***

- Super insulation (either vacuum panels or thicker walls) is likely to reduce loads.

- Dual loop (or staged) systems using two compressors may replace single compressor/evaporator designs to gain theoretical energy efficiency advantages.

4. ***Small compressors are inefficient because of a lack of economic incentive to develop efficient models; substantial improvements can be obtained by applying currently available technology.***

- Current small compressors are relatively inefficient, mainly due to inefficient motors and high levels of suction gas heating.

- Current technology exists to make efficient motors for small compressors at a reasonable price.

- It is technically feasible to develop and produce an efficient small compressor, having an EER greater than 5.0

5. ***Energy improvements from improved small compressor performance are possible with improved motors and application of other existing technology. Dual loop and super-insulation systems should be economically viable with this currently available (but as of yet) unmanufactured technology.***

- Small compressors can be improved to efficiency levels within 10 to 15 percent of large compressors for approximately $15.00; the energy gain they would produce in a dual loop system would be worth roughly $100.

- Research and improved manufacturing could improve the benefit/cost calculations used in this report.

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Acknowledgements

During the course of this work, a number of compressor, motor, and variable speed drive manufacturers were contacted, whose assistance and cooperation we acknowledge and appreciate:

- Compressors: Tecumseh Products Co., Copeland, Americold, Danfoss, Embraco, Sanyo, and Matsushita
- Motors: General Electric, A. O. Smith, Emerson, and Baldor
- Variable speed drives: Toshiba, Emerson, Mitsubishi, Hitachi, Westinghouse, Magnetek, Lenze, Vee Arc, Ranco, Inland, PMI, Minarik, EG&G, Fasco, Boston Gear, and Graham

1.0 Introduction

The design of the domestic refrigerator-freezer will be undergoing a significant set of changes over the next several years, driven by interrelated developments in a number of national energy and environmental policy areas - the CFC phase out, growing concerns about global warming, DOE appliance energy efficiency standard setting, and others. These developments have created the need for a comprehensive examination of the design options for domestic refrigerator/freezers. These interrelated options include compressors, motors, refrigerants and lubricants, the refrigeration cycle, cabinet insulation, and other aspects of cabinet thermal design.

This report covers the technology status of the small hermetic compressors used in the refrigeration system of domestic refrigerators, addressing the potential for and cost of improvements in the efficiency of small hermetic refrigeration compressors.

The environmental and energy policy background is discussed briefly below.

1.1 Stratospheric Ozone Depletion and the CFC Phase Out

Evidence accumulated over the past 15 years indicates that fully halogenated chlorofluorocarbons (CFCs) have caused measurable deterioration of the atmosphere's stratospheric ozone layer, which plays a significant role in attenuating solar ultraviolet radiation. Increased levels of ultraviolet radiation would have a large number of undesirable effects, including increased levels of skin cancers. Over the past few years, this subject area has received renewed attention as the result of observations in the mid-1980s of "gaps" in the ozone layer in the vicinity of the poles. As a result of this attention, the Montreal CFC protocols were concluded in Fall, 1987. This international agreement was signed and ratified by the major free world industrial nations requiring a freeze, then phased production curtailments, of CFCs. By 1998, production of CFC-11, CFC-12, CFC-113, CFC-114, and CFC-115, as well as certain "halons" were to be reduced to 50% of 1986 levels. An additional provision provided for periodic review of scientific evidence and adjustment of allowable levels of production accordingly. The reassessment completed in 1990 resulted in a nearly total phase out of CFCs by the year 2000. The Clean Air Act of 1990 has codified this accelerated CFC phase out schedule into U.S. environmental law. In April of 1991, NASA reported the results of satellite-based measurements of stratospheric ozone levels indicating that ozone depletion of 5% over the mid latitudes has already occurred. The result of this development has been further acceleration of the timetable for CFC phase out. In November, 1992, the Montreal Protocol Copenhagen Amendments accelerated the complete phase out of CFCs to January 1, 1996. This has a direct impact on R/F insulation and compressors which have been designed around the characteristics of CFC-11 and CFC-12, respectively.

1.2 Global Warming

Concurrent with the accumulation of scientific evidence of stratospheric ozone depletion, increasing concerns have been developing about global warming caused by increasing atmospheric concentrations of carbon dioxide and the trace greenhouse gases. The most significant of the trace greenhouse gases are the CFCs and methane.

The fact that the CFCs are powerful greenhouse gases has reinforced the pressures to accelerate the CFC phase out time table. Measures to limit or reduce CO_2 emissions have been proposed. Because CO_2 is one of the basic combustion products of all of the fossil fuels used to produce energy for heating, transportation, and electric power generation, measures to reduce CO_2 emissions require the burning of less fossil fuels. One regulatory measure to bring this about is increasing energy efficiency standards.

1.3 DOE Appliance Energy Efficiency Standard Setting

In February, 1989, DOE issued a final rulemaking under the Energy Policy and Conservation Act, as amended, establishing minimum energy efficiency standards for most categories of consumer appliances (Federal Register, 1989a). Depending on the category, the standards take effect between 1990 and 1993. The standards for domestic refrigerators and freezers went into effect on January 1, 1990, generally requiring efficiency levels in line with the most efficient products available in the late 1980s, whose efficiency was nearly double the levels prevailing only 10 to 15 years earlier.

Global warming (and national energy security issues) have resulted in the recent adoption of significantly reduced levels of allowable electric energy consumption for all categories of domestic refrigerators and freezers, effective on January 1, 1993 (the 1993 standards reduce allowable energy consumption by approximately 30% from the levels under the current Federal regulations that took effect on January 1, 1990) (Federal Register, 1989b).

1.4 Other Regulatory and Policy Initiatives

States are instituting reforms in planning and rate making that put demand reductions on an equal playing field with building additional supply capacity. Integrated resource planning, adopted by many states, requires utilities to evaluate every "resource" (demand reduction or supply) in terms of total societal cost.

California, Oregon, Washington, most of the New England states, Wisconsin, and New York have adopted ratemaking processes in which utility rates or return on investment is adjusted so that utilities do not lose profits for forgone Kwh sales, but can profit from demand reductions. In California and several other states, the non-pollution aspect of demand reductions has led regulators to allow shared savings of customer bill reductions to further increase utility profits. As a consequence of this change in utility regulation, the demand for efficient refrigerators is rising.

The Golden Carrot/early retirement program is one concrete manifestation of this trend. Under the Golden Carrot, utilities are banding together to pool rebates to produce an incentive for production of a R/F that is 30% better than DOE's 1993 standard in the 18 to 22 cubic foot range. With the impetus described earlier from CFCs, global warming, and other state regulatory reforms, the Golden Carnot will provide a strong incentive for vast improvements in energy efficiency.

2.0 Issues in Compressor and Motor Selection

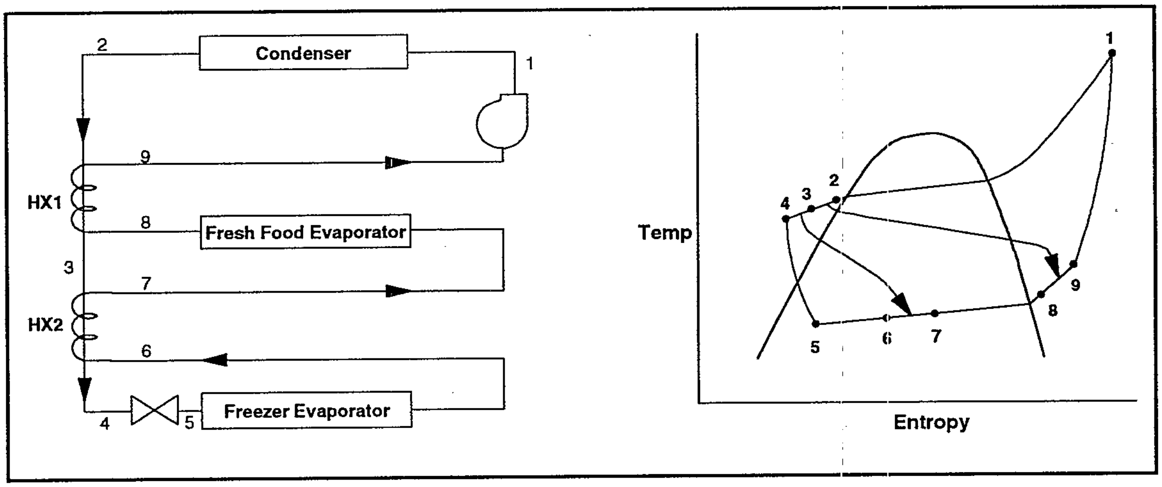
To meet the dual challenge of designing for new blowing agents and refrigerants, while meeting much higher efficiency standards, significant changes in the design of both the cabinet and the refrigeration cycle are under consideration. There are many design options, and it is important to evaluate the potential of each option to contribute significant energy savings, without using CFCs and without increasing the cost of the refrigerator beyond the value added by the change (with respect to energy costs or utility) or decreasing the utility of the refrigerator to the consumer. Design options that are being pursued in current R&D programs include:

- Low thermal loss cabinets ("super-insulated boxes") reduce energy consumption by reducing the amount of cooling that is needed. To utilize heat exchangers effectively and minimize cycling losses, the compressor capacity should be reduced in proportion to the thermal load. However, sufficient compressor capacity may still be required to provide a sufficiently fast pulldown for food preservation.

- Dual refrigeration loops (separate refrigeration systems for the refrigerator and freezer compartment) take advantage of the increased COP at the higher evaporator temperature that can provide the required cooling of the fresh food compartment.

- The Lorenz cycle, shown schematically in Figure 2-1 uses a non-azeotropic refrigerant mixture with two evaporators and an interchanger to operate the fresh food compartment and the freezer at separate evaporator temperatures for higher efficiency. (Lorenz, 1975)

Figure 2-1: Lorenz-Meutzner Refrigerator/Freezer Cycle



- Variable speed compressor operation can save energy by allowing continuous operation at low capacity, eliminating cycling losses and allowing more efficient utilization of heat exchangers. Overspeed operation can provide additional capacity for pulldown. The latter characteristic might be particularly advantageous with high performance cabinets.

One consequence of these design changes is a decrease in the compressor capacity required to best match a given size refrigerator. Improved insulation and cabinet design will reduce cooling loads, reducing required compressor capacity. Two compressor/two evaporator systems meet the freezer and fresh food compartment heat loads with separate refrigeration systems, obtaining, in theory, significant improvements in the efficiency with which the load in the fresh food compartment is met (higher evaporating temperatures, no defrost cycle). Even less compressor capacity is needed, especially in the fresh food compartment where nominal compressor capacities of only 200 Btu/hr may be needed. While the Lorenz cycle does not inherently result in a drastic compressor capacity reduction, other compressor problems, such as high starting torque requirements, have been observed. Variable speed compressors will tend to be smaller displacement, with pulldown requirements met by overspeed operation.

Present commercially available low capacity compressors (<600 Btu/hr nominal capacity) have very poor efficiencies, low enough in some cases to completely negate the gains obtained from the design options described above. To realize the efficiency benefits of reduced loads and dual evaporator systems will require improved efficiency, lower capacity compressors.

Regardless of the R/F cabinet and refrigeration cycle design approach taken, increases in the efficiency level that is available in refrigerant compressors will result in proportional increases in the efficiency of the refrigerator using the compressor.

A major issue for compressor design is the change in refrigerant from CFC-12 to a low ozone depletion, low global warming potential refrigerant. While CFC-12 has been shown to be a significant part of the cause of both stratospheric ozone depletion and global warming, it is an excellent working fluid for domestic refrigerator/freezers. It has a favorable pressure-temperature relationship, good thermodynamic efficiency, stability, total miscibility with low cost mineral oil, and moderate temperature rise with compression and is non-flammable, non-toxic, and low cost. Alternate refrigerants that do not contain chlorine or have currently acceptable ozone depletion potentials do not possess identical attributes of CFC-12. Thus, compressor modifications or new designs will be required to adapt to the characteristics of a selected alternative refrigerant. Table 2-1 lists some of the potential alternative refrigerants and their status, including potential substitutes for CFC-11, CFC-114, and CFC-502, as well as for CFC-12.

The major working fluid options include the near drop in replacements for CFC-12 (i.e., those refrigerants having vapor pressure-temperature curves close to that of CFC-12), lower vapor pressure refrigerants such as HCFC-124, and non-azeotropic refrigeration mixtures (NARMs). Lower vapor pressure refrigerants might be utilized in low capacity systems (design options described above), if shown to result in higher efficiency of low capacity compressors, by virtue of the larger displacement that would be needed. The Lorenz cycle would utilize a NARM.

Table 2-1: Alternative Refrigerants

Substitute Refrigerant	Displaced Refrigerant	Probable Availability	Description, Status, Comment
HCFC-22	CFC-12 CFC-502	Current	Commercially available, widely used refrigerant. Contains chlorine, will be phased out under Montreal Protocol Copenhagen Amendments
HFC-134a	CFC-12	Current	Commercially available, rapidly expanding production
Ternary	CFC-12	(w/HCFC-124) 1993	Available in limited amounts
HFC-152a	CFC-12	Current	Commercially produced and sold in fairly small quantities, used primarily as a component in CFC-500 (26%) and as a component in aerosol propellant blends
HFC-123	CFC-11	1990-1991	Toxicity tests have shown sufficient toxicity to set AEL at 10 ppm; commercially available
HFC-124	CFC-114	1993-95	Co-product of HCFC-123 production. Long term toxicity testing started
HFC-125	CFC-502	Available in limited amounts	Near-term availability in blends to replace CFC-502
HCFC-141b	CFC-11	Current	Commercial production began in July 1988. Toxicity testing underway. Possible use as a foam blowing agent
HCFC-142b	CFC-12, 114	Current	Used in R22/R142b blends
NH_3	CFC-11 CFC-12 CFC-502	Current	Commercially available. Widely used in industrial refrigeration sector. Toxic with low flammability

Source: Arthur D. Little, 1993

The major issues that need to be considered in adapting the compressor design to an alternate refrigerant include the displacement required to obtain the intended capacity, lubricant selection, and material compatibility, especially the motor winding insulation.

In summary, higher efficiency compressors are needed, especially in smaller capacities. Motor technology is an important consideration, because increasing the efficiency of the compressor motor is a straightforward way to improve compressor efficiency. For variable speed compressors, the variable speed motor and electronic drive represent the major technology component and the major cost driver.

2.1 Compressor Technology Survey

In view of the importance of compressor performance to the potential efficiency improvements of various design alternatives, this review of the technology of small hermetic refrigeration compressors was undertaken as part of a larger study to evaluate the options for maximizing the efficiency of domestic refrigerators. The compressor characteristics documented in the report are being used in the overall study as part of the input database for modeling and evaluating design options.

This report covers the technology status of small hermetic compressors used in the refrigeration system of domestic refrigerators, addressing the potential for and cost of improvements in the efficiency of small hermetic refrigeration compressors, with particular emphasis on smaller (<800 Btu/hr nominal capacity) compressors.

This report is intended to serve three functions: 1) description of the state-of-the-art of current compressors and the potential for future improvements; 2) summary of performance and cost data as input to evaluations of refrigerator/freezer system design options; and 3) present preliminary results indicating the level of R/F energy consumption reductions that can be obtained through the use of high efficiency small compressors.

3.0 Compressor Technology: Current State of the Art

3.1 Description of Current Compressor Technology

Two types of compressors are used in current domestic R/F units, in the United States and worldwide:

- Reciprocating
- "Rotary" (Rolling Piston - stationary vane)

Both types are welded hermetic, i.e. the compressor pump and its motor are sealed inside a welded shell. The refrigerant gas connections are welded to the shell and the electrical connections to the motor are made through an insulated pass through bonded to the metal shell. This hermetic arrangement prevents the loss of oil and refrigerant which could occur through rotating seals or mechanical fittings.

3.1.1 Reciprocating Compressors

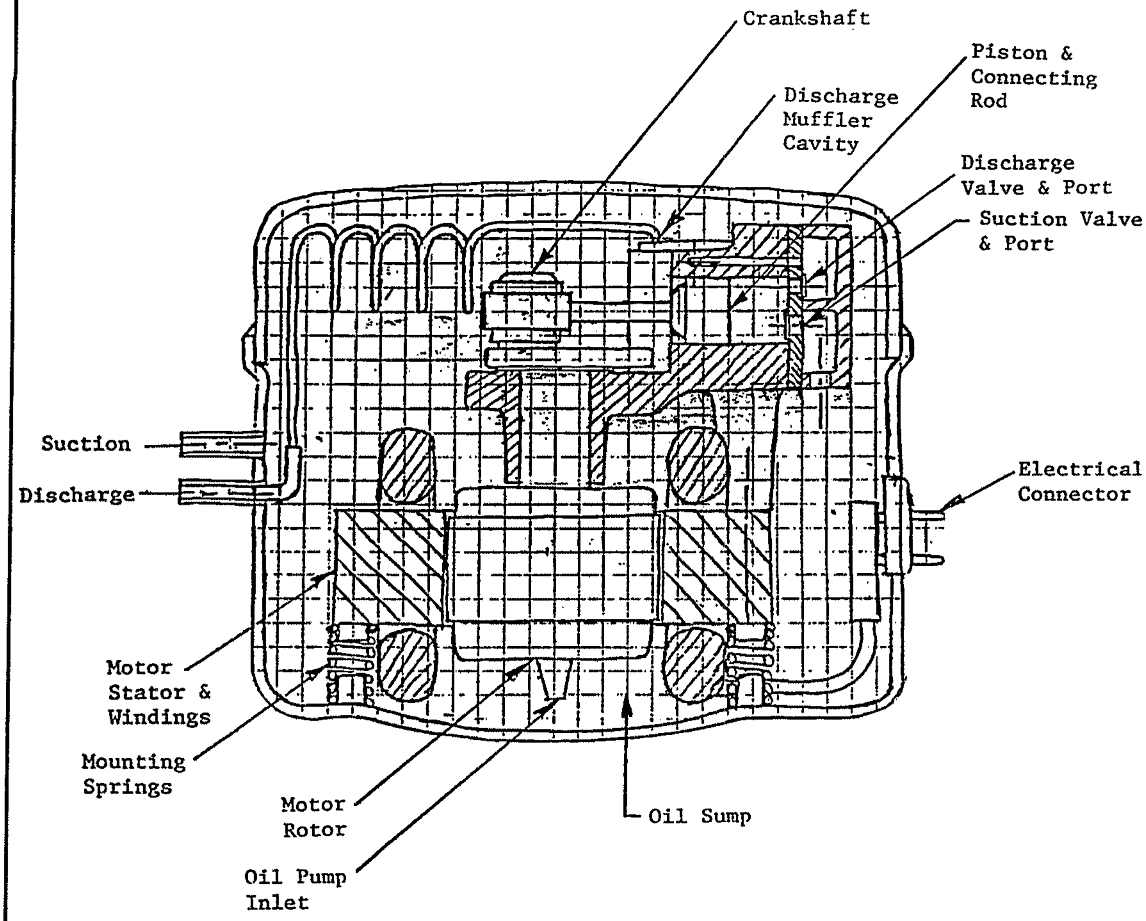
The reciprocating compressor has been and still is the most common type used. It has reached a high state of development, is mechanically efficient and reliable and is relatively less expensive to manufacture than its alternative, the rotary, because of its lower overall sensitivity to manufacturing tolerances.

A typical reciprocating compressor used in domestic R/Fs (Figure 3-1) is a single cylinder device with a piston driven by a crankshaft which is an integral extension of the driving motor shaft. The piston is connected to the crank by a connecting rod and wrist pin. The piston reciprocates in a stationary cylinder secured to the motor stator. A cylinder head attached to the cylinder houses two reed valves, one of which, the suction, opens into the cylinder and the other, the discharge opens outwardly from the cylinder. An oil pump is located in the non-driving end of the motor shaft which supplies oil to the rotating and reciprocating parts of the pump and motor.

Gas is drawn from the R/F evaporator into the compressor shell in the space in the cannister surrounding the motor/pump combination. It circulates within the space aided by the fan effect of the motor rotor. This cools the motor and the pump. A significant portion of the heat picked up by the gas is convected to the shell for dissipation to ambient air. The heated gas is then drawn through the pump suction valve into the cylinder where it is compressed and ejected through the discharge valve and piped to the discharge connection on the shell.

Higher efficiency compressors may provide for a directed suction path from the inlet at the shell directly into a muffler assembly, reducing the superheating of the suction gas. Forced air cooling of the compressor shell may then be required to provide for motor cooling. The path of the suction gas ensures that oil which is discharged to the external system with the compressed gas is ultimately returned to the cannister and oil pump. It also allows liquid refrigerant which accumulates on shut down in the evaporator to be slugged into the cannister on start up rather than into the cylinder. This could damage the suction valve on the compression stroke. The liquid is vaporized harmlessly by the

Figure 3-1: Typical Small Hermetic Reciprocating Motor-Compressor



Source: ADL sketch of Danfoss IL3A Compressor (315 Btu/hr, 3.5 EER) heat dissipated into the suction gas and returned to the cycle (a small energy penalty, contributing to cycling losses, is incurred due to the compression work associated with liquid being vaporized in the crank case).

The pump/motor assembly is mounted in the shell with isolation springs to minimize vibration transfer to the shell. Mufflers contained within the shell as a part of the pump suction and discharge assemblies dampen gas pulsations.

3.1.2 Rotary Compressors

This class of compressors for R/F use in the range of interest (200 to 800 Btus) is limited to the rolling piston - stationary vane type rotary compressor. Rotary vane compressors were used in the past but are less efficient than either the reciprocating or rolling piston compressors and are no longer employed.

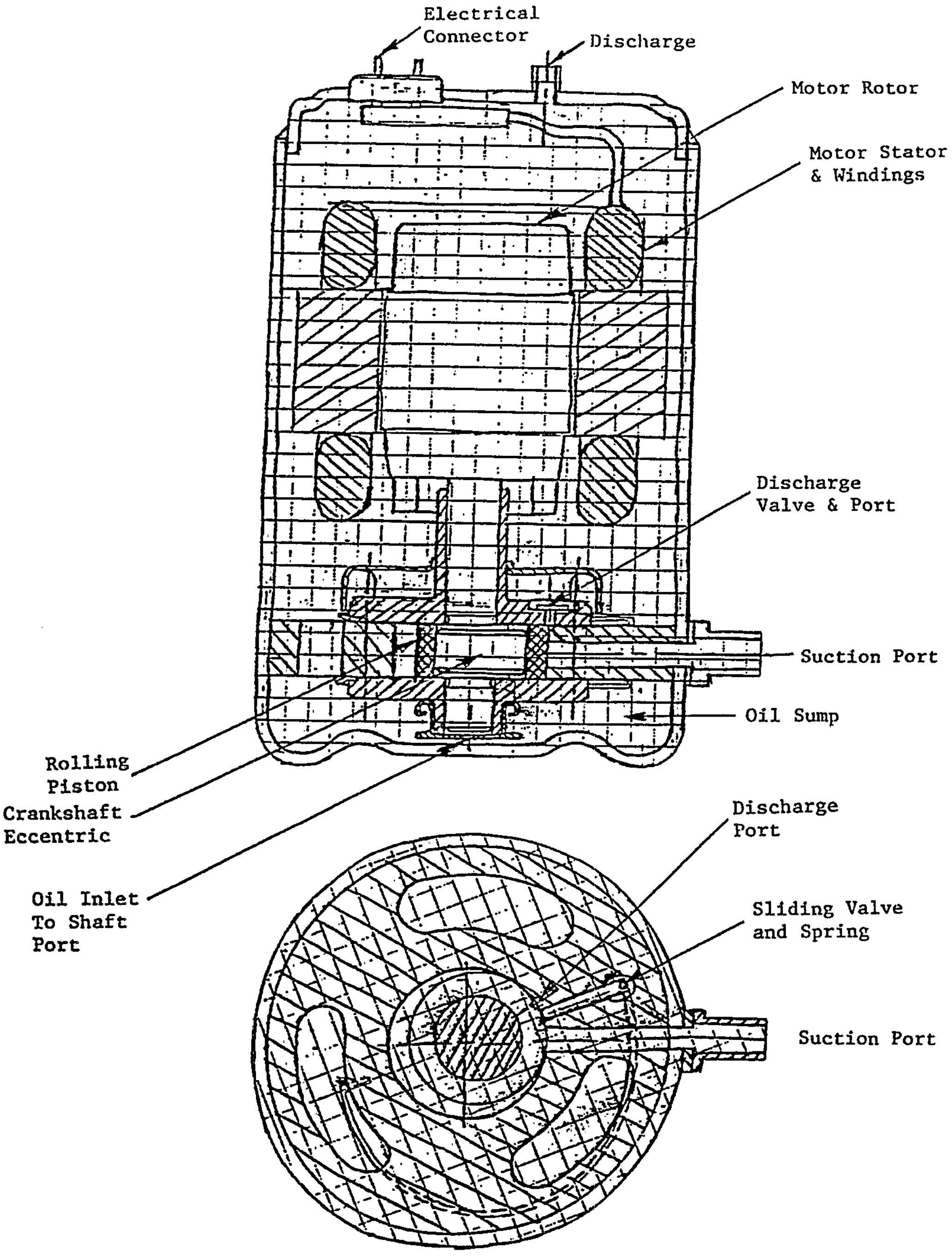
The rolling piston compressor (Figure 3-2) consists of a cylindrical "piston" rolling on the wall of a "cylinder" which is capped at both ends. The piston is driven in the circular orbit defined by its rolling on the cylinder wall, by an eccentric which is an integral extension of the motor shaft. A reciprocating vane located in the cylinder between the discharge and suction ports is held against the piston by a spring to provide a seal between the suction and compression sides of the piston. A reed valve is used on the discharge port to prevent back flow from the discharge line into the compression cavity and short circuiting from the discharge to the suction during the interval when the piston is between the discharge and suction ports. The former ensures variable pressure compression. Rotary compressors used in domestic R/Fs contain the motor and pump housed in a welded shell hermetically sealed against refrigerant leakage.

Suction gas is piped directly to the suction port which minimizes superheating. The suction gas is not used to cool the motor/pump combination. Because of the close clearances, rotary compressors are very sensitive to ingestion of both particulates and liquid slugs. To protect against both of these, all rotary compressors include a suction accumulator and strainer close coupled to the inlet.

Discharge gas is ported directly into the shell which is at discharge pressure (high side) where entrained oil has an opportunity to partially separate before the gas exits the discharge connection. A portion of the heat dissipated in the motor is transferred to the discharge gas. The motor stator is pressed into the shell which promotes conduction of heat to the shell. Oil is metered to the bearings from the sump through an orifice by the pressure difference from the shell side and the crank/piston bearing area which is at a pressure intermediate to the suction and discharge pressure.

The piston does not actually contact the cylinder walls or end caps, but is separated from them by a thin oil film. Close clearances are necessary to prevent gas blow by. Sealing of the clearances is enhanced by lubricating oil filling the clearance. Rotary compressors generally have a suction accumulator/filter closely coupled to the inlet of the pump to absorb liquid refrigerant and oil slugs at start up and prevent particulates from entering the close clearances of the pump.

Figure 3-2: Typical Small Hermetic Rotary Motor-Compressor



Source: ADL sketch of GE 2T/27RV/RS-36 Compressor (approximately 1000 Btu/hr)

The rotary pump has less vibration than the reciprocating pump. Its piston has an eccentric orbit and the vane reciprocates; the eccentrically orbiting rolling piston is dynamically balanced by a counterweight and trim weight on the motor shaft. The reciprocating motion of the vane is not counterbalanced, but the product of the weight and stroke length of the vane is at least an order of magnitude less than that of the piston and connecting rod of a reciprocating compressor having the same capacity. The noise level (vibration in the audible range) of a rotary is higher than an equivalent reciprocating pump even though its total vibration is less, because the motor-pump assembly is rigidly attached to the compressor shell.

The principal disadvantage of this pump is that it requires close clearances between piston and cylinder, piston and crankshaft, crankshaft and support bearings and piston and cylinder caps to prevent excessive gas blow by. These constraints require close machining tolerances (on the order of fractions of a thousandth of an inch) over a much greater total surface area than the reciprocating pump with attendant cost penalties and make the pump less tolerant of wear.

There is a general expectation that the rolling piston compressor is not going to be as efficient as a reciprocating pump in the lower capacity ranges. This results from the inherently high surface to volume ratio of the cylinder and piston. This increases the leakage paths and decreases the mechanical efficiency at a greater rate with reducing size than it does for a reciprocating compressor. This opinion was supported by Professor Soedel of the Herrick Laboratories at Purdue University (reference 15) and extended to scroll compressors particularly because of the disproportionate increase in leakage paths.

The principal advantages of the rolling piston pump are that it can be a high side pump with attendant low suction gas superheat, high volumetric efficiency because of its inherently low clearance volume and unrestricted suction port. These advantages in energy efficiency are offset by higher levels of other losses, resulting in overall efficiencies being less than the best current reciprocating pumps in the higher capacities. Rotaries are more compact and lighter in weight than recips of comparable capacity; the inherent potential cost savings have been a major reason for their growth in market share during the 1980's.

3.1.3 Motor

Both types of hermetic compressors are driven by single phase squirrel cage induction motors.

The motors consist of pairs of stationary (stator) electromagnets placed 180 degrees apart, energized by "run" windings. The current produces an oscillating magnetic field in the plane centered on the motor axis. This field induces a current in rotor conductors which in turn generates a magnetic field in the rotor iron core. The rotor field opposes the stator field producing a torque in the rotor.

An induction motor depends upon the rotor magnetic field being generated by the induced rotor current. This current is induced only when there is a speed difference between the oscillating stator and rotating rotor fields.

The speed of the stator field is determined by the alternating current frequency and the number of stator coil pairs; e.g., a 60 hertz current with a two coil (pole) stator produces a magnetic field oscillating at 3600 rpm; a four pole motor at the same frequency produces an 1800 rpm field speed. The motor speed is the speed of the rotor, which must rotate at a speed less than that of the stator field. A speed difference, or "slip", of approximately 3%, corresponds to the rated torque and power output of the motor. For a two pole motor this speed is about 3500 rpm.

Both the stator and rotor magnets are made of laminated iron. The stator coils are generally of wound copper wire. The rotor coil is formed of aluminum bars cast into passages in the iron core resembling a squirrel cage configuration, thus the name "squirrel cage" motor. Both magnetic fields induce eddy currents in the iron resulting in heating losses. These are minimized by laminating the iron.

When the rotor is stationary, the oscillating stator magnetic field produces a balanced force in the rotor with no net torque, consequently some assistance is required to start a single phase motor. This is accomplished by a pair of electromagnetic poles located 90 degrees from the "run" magnets called the "starting windings". On starting, these windings in conjunction with the "run" windings produce a rotating magnetic field which induces a net torque in the rotor. Once the rotor has achieved a self sustaining speed the start circuit is switched out of the circuit.

A resistor or a capacitor is employed in series with the start winding to enhance the phase difference and thus the starting torque. The windings are switched out of the circuit by a mechanical relay or a positive temperature coefficient resistor (PTCR). The latter upon heating increases in resistance to reduce the current flow to negligible proportions.

A motor with a starting winding only is called a split phase motor. A motor with a resistor in series with the start winding is called a Resistor Start Induction Run (RSIR) motor. These types produce a non-uniform rotating field which cause an objectionable vibration in the rotor and must be switched out of the circuit after starting.

If a capacitor is connected in series with the start winding a 90° phase difference between the run and start windings occurs resulting in a greater starting torque. This type of motor is called a Capacitor Start (CS) motor.

A capacitor smaller than the starting capacitor can be inserted into the starting winding circuit and the starting winding then operated continuously. This results in a higher power factor which in turn reduces motor current, decreasing winding resistance losses and improving motor efficiency by about 10%. This is called a capacitor run (CR) motor. If the motor also employs an additional capacitor in parallel with the "run" capacitor for starting the combination is called a "Permanent Split Capacitor" (PSC) motor. The "starting" capacitor is switched out of the circuit after starting.

The winding resistance, eddy currents and windage friction represent the losses of a motor. These produce heat which is transferred to the refrigerant gas and then to the shell for dissipation to the ambient air.

Currently, larger (>700 Btu/hr), higher efficiency compressors for domestic R/Fs use PSC motors. Smaller compressors generally use lower cost RSIR motors.

3.2 Performance

As discussed in Section 2, it is likely that compressors for future R/F designs will be smaller than those current employed, approximately one half to one third of current averages:

- Present capacity range - 500 to 1500 Btu/hr
- Probable future capacity range - 200 to 800 Btu/hr

A partial survey of the major U.S. refrigerator/freezer manufacturers and their compressor suppliers (Table 3-1) shows that a wide selection of models in the 200 to 800 Btu/hr range is available to U.S. R/F manufacturers (Table 3-2). As shown by the data in Table 3-2 and graphically in Figure 3-3, an efficiency decrease parallels the decrease in size, as indicated by the EER* value.

In Figure 3-3, several curves are shown to fit the individual EER data points. Somewhat arbitrarily, below 600 Btu/hr is labeled "small compressor", above is labeled "large compressor." Current production state of the art represents a curve fit of the values of compressors that are in full scale production and use by OEMs. Near term SOA is a curve fit of the best EER values for compressors we could identify for which samples are being supplied to OEMs, with production likely to occur as needed for 1993 appliance production.

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* EER (Energy Efficiency Ratio) is defined as the compressor refrigeration effect in Btu per hour divided by the compressor motor input in Watts (combined units: Btu/Watt-hr).

This expression is a variation of the "Coefficient of Performance" (COP) but well adapted to the units in general use for expressing compressor data. The standard conditions for measuring the terms of this value are -10°F saturated suction temperature, 130°F saturated discharge temperature, 90°F liquid and suction vapor temperatures and 90°F ambient temperature.

Table 3-1: Refrigerator/Freezer Manufacturers and Compressor Suppliers

R/F Manufacturers	Compressor Suppliers
General Electric/Hotpoint	General Electric Danfoss Tecumseh
Whirlpool	Panasonic (Matsushita) Embraco Aspera
White/Frigidaire	Americold
Amana	Panasonic Sanyo Tecumseh
Admiral/Maytag	Panasonic Sanyo Embraco Tecumseh

Source: References 4, 24

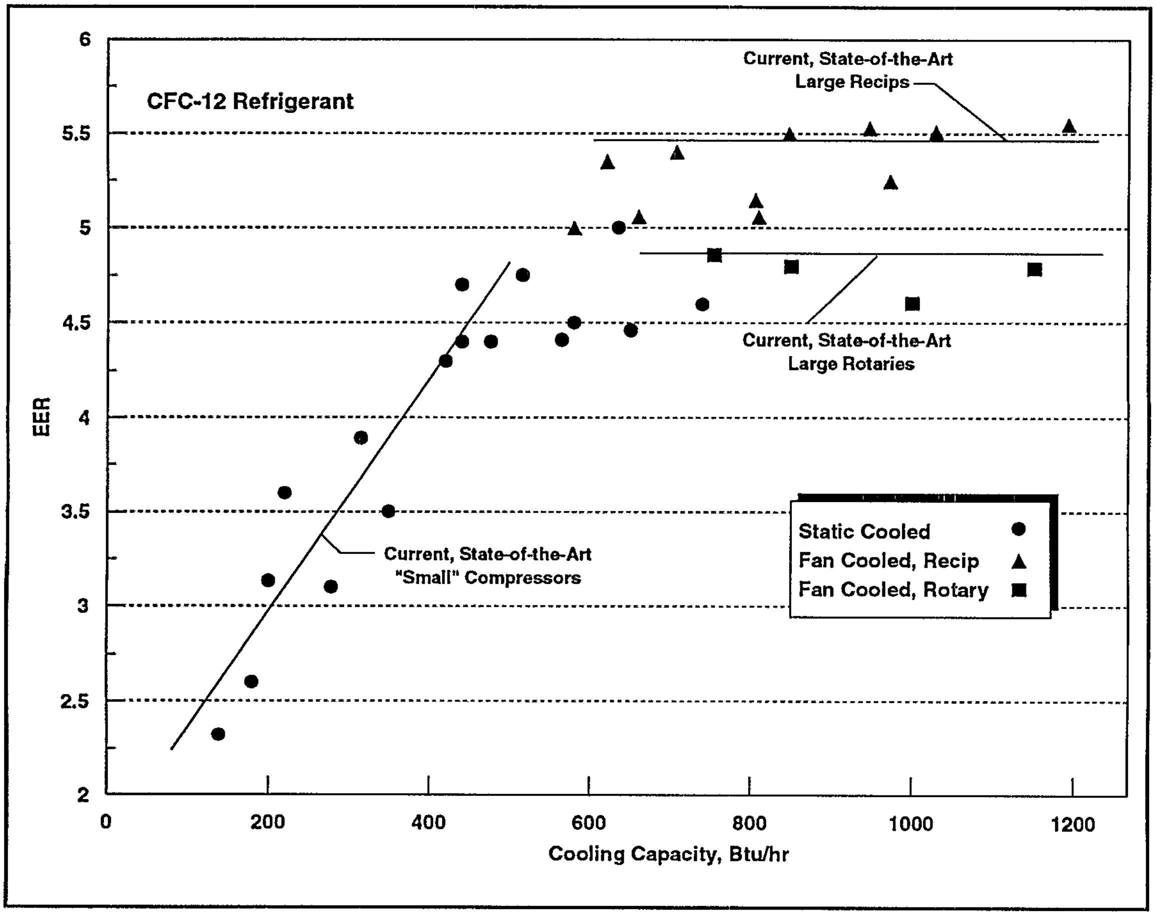
Table 3-2: Partial List of Current High Efficiency Refrigerator/Freezer Compressors

Manufacturer	Cap	Dspl	EER	Motor	Cooling	Type
Sanyo (CQ30)	139	0.109	2.32	RSIR	Static	Recip
Sanyo (C-M40L12C)	180	0.13	2.6	RSIR	Static	Recip
Tecumseh (AZ132OD)	200	0.136	3.13	RSIR	Static	Recip
Embraco (EMI 20ER)	220	0.139	3.6	RSIR	Static	Recip
Panasonic (S070LKAA)	278	0.184	3.1	RSIR	Static	Recip
Embraco (EM 30SC)	315	0.184	3.89	RSCR	Static	Recip
Panasonic (S090LKAA)	349	0.217	3.5	RSIR	Static	Recip
Embraco (EM 40SC)	420	0.23	4.3	RSIR	Static	Recip
Americold (ST104)	440	0.217	4.7	PTCR/CR	Fan	Recip
Embraco (EMI 40ER)	440	0.23	4.4	RSIR	Static	Recip
Panasonic (D112LRAA)	476	0.264	4.4	RSCR	Static	Recip
Panasonic (DA43L67)	516	0.264	4.75	RSCR	Static	Recip
Embraco (EM 55SC)	565	0.305	4.41	RSCR	Static	Recip
Embraco (EMI 55ER)	580	0.305	4.5	RSIR	Static	Recip
Americold (ST105)	580	0.277	5	PTCR/CR	Fan	Recip
Americold (HG106-1)	621	0.277	5.35	PTCR/CR	Fan	Recip
Panasonic (DA51L88R)	635	0.311	5	RSCR	Static	Recip
Panasonic (FN40R80R)	651	0.248	4.46	RSCR	Static	Rotary
Americold (ST106)	660	0.312	5.06	PTCR/CR	Fan	Recip
Americold (HG107-1)	708	0.312	5.4	PTCR/CR	Fan	Recip
Tecumseh (AE1370W)	740	0.421	4.6	PTCR/CR	Static	Recip
Panasonic (RA48L83R)	754	0.294	4.86	RSCR	Fan	Rotary
Panasonic (DA66L11R)	806	0.402	5.15	RSCR	Fan	Recip
Americold (ST107)	810	0.366	5.06	PTCR/CR	Fan	Recip
Americold (HG108-1)	848	0.361	5.5	PTCR/CR	Fan	Recip
Panasonic (RA53L11R)	850	0.323	4.8	RSCR	Fan	Rotary
Americold (HG109-1)	947	0.401	5.53	PTCR/CR	Fan	Recip
Panasonic (DA73L13R)	973	0.469	5.25	RSCR	Fan	Recip
Panasonic (FN60R12R)	1000	0.372	4.61	RSCR	Fan	Rotary
Americold (HG110-1)	1029	0.473	5.51	PTCR/CR	Fan	Recip
Panasonic (FN70R16R)	1151	0.435	4.79	RSCR	Fan	Rotary
Americold (HG111-1)	1193	0.5	5.55	PTCR/CR	Fan	Recip

CAP — Capacity (Btu/hr)
DSPL — Displacement (cubic inches)
RECIP — Reciprocating
RSIR — Resistance Start Induction Run
PTCR/CR — Positive Temperature Coefficient Resistor Start/Capacitor Run
RSCR — Resistance Start Capacitor Run
PSC — Permanent Split Capacitor

Source: Engineering data supplied by the manufacturer of each model

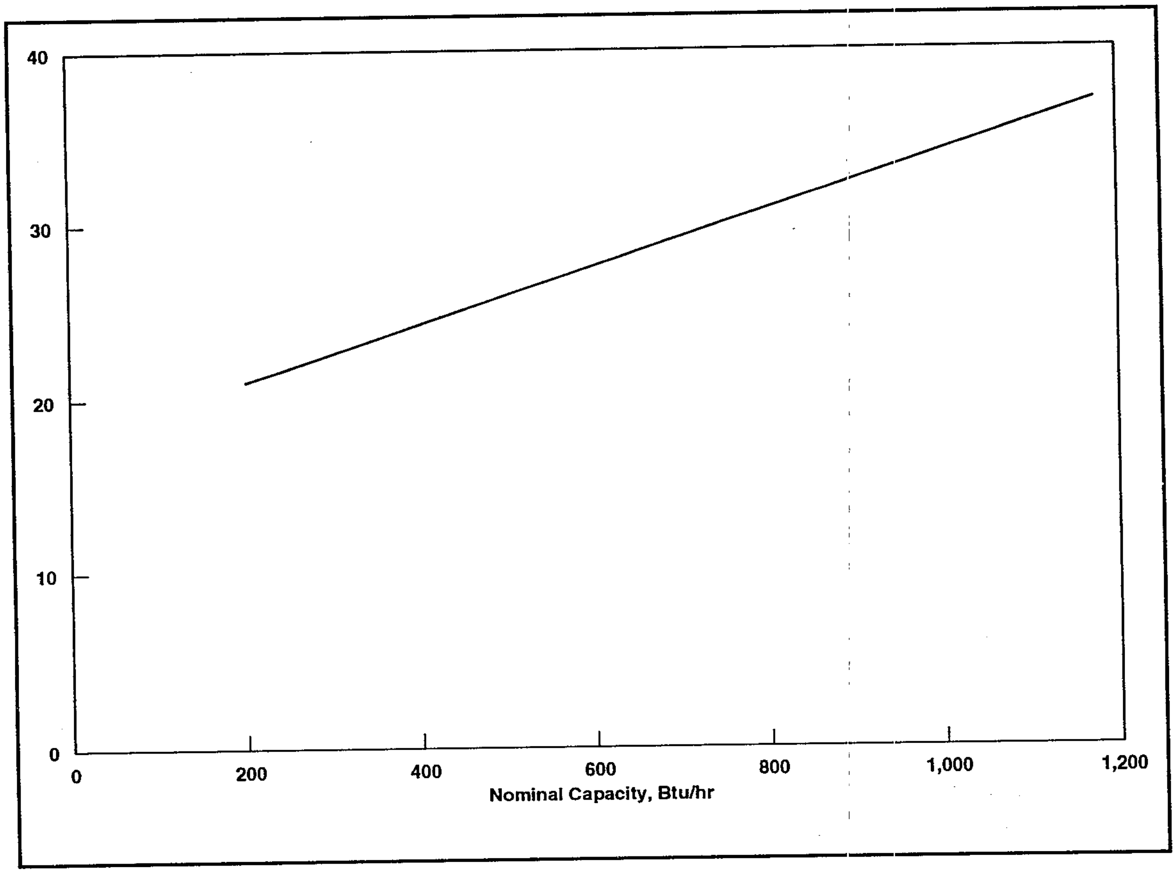
Figure 3-3: **Compressor Efficiencies versus Capacity**



3.3 Compressor Costs

When sold on an OEM basis, refrigeration compressor prices are negotiated between the compressor manufacturer and the refrigerator manufacturer. Compressor manufacturers consider negotiated OEM pricing to be commercially/competitively sensitive information, and are, therefore, reluctant to discuss the subject in detail. Figure 3-4 plots the estimated OEM price level versus capacity corresponding to the "average efficiency" versus capacity plotted in Figure 3-1, based on general discussions of OEM price levels with several manufacturers.

Figure 3-4: Estimated Price of Average Efficiency Refrigeration Compressors

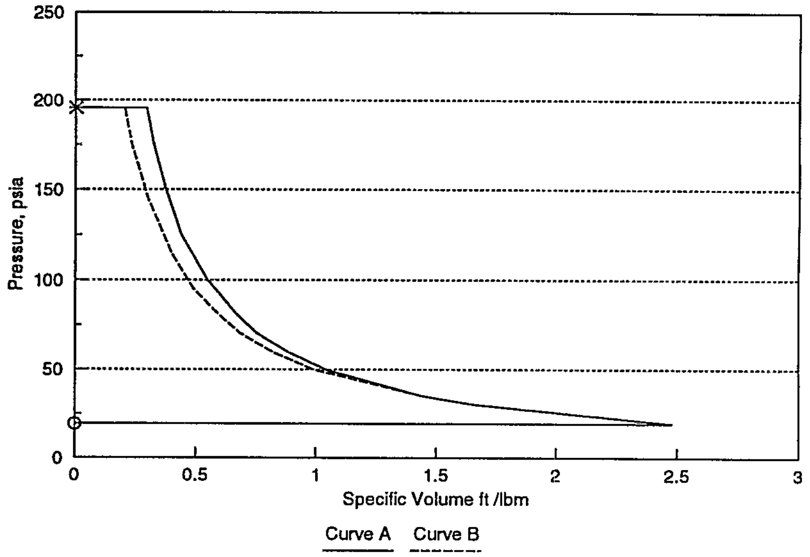


3.4 Losses

As discussed above, the maximum efficiency level that is available in current, production compressors, in terms of standard rating conditions EER, is 5.0 Btu/Watt-hr for "large" (nominal capacity above 700 Btu/hr) and 3.5 for the small (200 Btu/hr) compressors of specific interest to this study. The 5.0 EER of large compressors represents a significant improvement over the efficiency levels that were available in 1980; much smaller improvements were made to the smaller compressors over this time period. To place the following discussion of losses, and the discussion of potential improvements (Sections 4 and 5) in perspective, it is instructive to consider the thermodynamic limits on compressor EER.

At the standard rating conditions, two thermodynamically limiting compression processes can be defined that are *theoretically* consistent with the rating conditions. Figure 3-5 plots the two processes, for CFC-12.

Figure 3-5: **Ideal PV Processes at Standard Conditions**

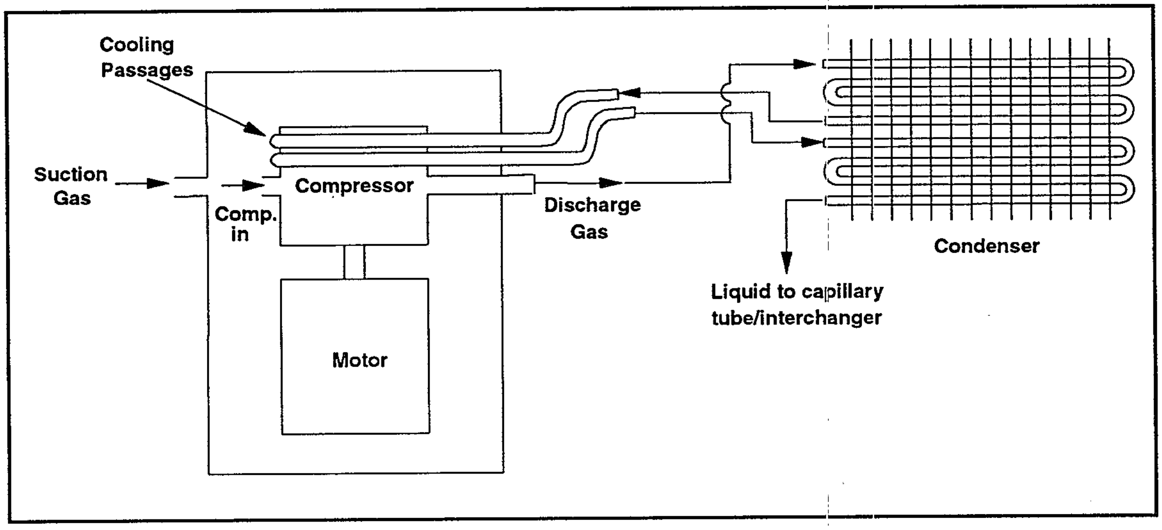


Curve A: Reversible adiabatic (isentropic) compression from the suction conditions (-10°F saturation pressure of 19.19 psia for R12, superheated to 90°F) to the discharge pressure (130°F saturation pressure 195.71 psia for R12). Compressor efficiencies are most commonly quoted relative to this thermodynamically "ideal" case. Implicit in the selection of an adiabatic process as the standard is the presumption that heat rejection directly from the compression process is impractical. For this process with CFC-12, the ideal EER is 9.26, at 90°F suction temperature -10°F saturated suction temperature, 130°F saturated discharge temperature, and 90°F liquid and ambient air temperature.

Curve B: Reversible adiabatic compression from the suction conditions to 130°F, followed by reversible, isothermal compression to the discharge pressure. As indicated by the curve, isothermalization reduces the compression work; the resulting ideal EER is 9.8. Conceptually, isothermalization of the compression process to this extent might be approached by routing partially condensed refrigerant from the condenser to passages in the cylinder body and head, then returning the refrigerant to the condenser, as shown schematically in Figure 3-6.

The difference in performance between these two cases is relatively modest. While the compression process of domestic refrigerator compressors is, in fact, not at all adiabatic, with internal heat transfer and heat rejection from the compressor shell to ambient air having a significant effect on the process, these heat transfer processes result in suction gas heating, as well as isothermalization. For the purposes of the discussions in this section and Sections 4 and 5, isentropic compression is taken as the "ideal" process. Figure 3-5 was developed for CFC-12, the refrigerant that is the basis for all *current*

Figure 3-6: Conceptual Means of Isothermalizing the Compression Process



compressors. Table 3-3 summarizes the ideal EER for CFC-12, HFC-134a, HFC-152a, and cyclopropane; at standard conditions the difference in ideal EER between these fluids is small.

Table 3-3: EER Limiting Cases, Based on Alternate Compression Processes

	Ideal EER*	
Refrigerant	Reversible Adiabatic	Isothermal** 130°F
CFC-12	9.26	9.8
HFC-134a	9.32	9.8
HFC-152a	9.26	9.8
Cyclopropane	9.15	9.7

* At standard rating conditions, 100% efficient motor and thermodynamically reversible compression process.

** Curve B in Figure 4-1

Table 3-4 presents, for several compressor configurations, an approximate breakdown of the electric input power among the basic categories of losses and useful (basis: reversible adiabatic compression) refrigerant vapor compression. The "high efficiency reciprocating" compressor represents the highest efficiency compressors that are currently available in production, in nominal capacities above 750 Btu/hr. Compressors having this level of efficiency have been in commercial production for the past 2 or 3 years, and are the result of development efforts that were initiated in the late 1970s. The "small, low efficiency reciprocating" compressor is representative of current compressors whose nominal capacity is less than 400 Btu/hr. The "typical rotary" is representative of both Japanese and GE rotary compressors, available in nominal capacities greater than 600 Btu/hr. The following subsections discuss the individual loss mechanisms in greater detail. It appears that by employing techniques that are now used in larger compressors, efficiencies of smaller compressors can be increased significantly. Sections 4.0 and 5.0 discuss the potential for improvement for "large" and "small" compressors, respectively.

Table 3-4: Typical Compressor Input Power Distribution

	High Efficiency Reciprocating (EER = 5.0)	Small, Low Efficiency Reciprocating (EER = 3.2)	Typical Rotary (EER = 4.7)
Input Electric Power	100%	100%	100%
Motor Loss	17%	28%	22%
Mechanical Losses	8%	10%	10%
Suction Gas Heating	12%	16%	3%
Discharge Port Losses	2%	2%	4%
Low Side Pressure Losses	4%	4%	2%
Compression and Expansion Losses	2%	4%	3%
Piston Blow By/Internal Leakage	1%	1%	5%
Power to Compressor - Gas Delivered**	54%	35%	51%

3.4.1 Motors

Table 3-5 shows the efficiencies currently attained with compressor motors now in use for the range of interest. It also shows a significant deterioration of efficiency that parallels size. In this case, the reasons are largely due to less attention to performance in smaller motors in the interest of low costs rather than an inherent trend to reduced efficiency. GE, for example, says that it is not currently developing higher efficiency motors for compressors below 600 Btu/hr because of lack of demand. Conversely, because European and Japanese R/Fs are smaller than US models, there is more emphasis on efficiency in smaller motors. In the US, aluminum wire is used in small motor windings, whereas the Europeans and Japanese use more efficient copper.

The possible efficiencies in Table 3-5 can be attained by the following improvements:

- All motors operated as Permanent Split Capacitor motors,
- Low eddy current and hysterisis loss steel through better chemistry and annealing,
- Additional material (larger motors), i.e., more copper, bigger magnets, and
- Different types of laminations in some motor designs.

** Based on reversible adiabatic compression of actual delivered mass flow rate from shell inlet conditions to discharge pressure.

Table 3-5: **Motor Efficiencies**

Compressor Rating	Motor HP	Today's Motor		Possible	
		Type	Efficiency	Type	Efficiency
200 Btu	1/16	RSIR	70-73%	PSC	84%
400 Btu	1/8	RSIR	73-76%	PSC	85%
600 Btu	1/6	RSIR&PSC	78-82%	PSC	86%
800 Btu	1/4	PSC	80-84%	PSC	86%

Legend
RSIR Resistance Start Induction Run
PSC Permanent Split Capacitor

These improvements could result in motor costs from 1.5 to 3 times present costs, an increase over current motor costs of approximately $10 to $25, according to General Electric sources. The largest increases would be in the lower output motors used with the smaller capacity compressors. These cost increases, while substantial, would still leave the cost of smaller motors less than that of the current larger, more efficient sizes. The projected efficiencies show an essentially level curve for the range of interest. This effect alone would play a major role in flattening the compressor EER versus capacity curve.

A separate report on the technology of both constant and variable speed motors (reference 3) has been prepared under this project.

3.4.2 Losses Within the Pump

The major losses in the pumps (i.e., the compressor less the motor) are mechanical friction losses, including bearings and cylinder/piston friction, pressure losses through mufflers, valve ports and reeds, other suction pressure losses, and suction gas superheating.

3.4.2.1 Mechanical Efficiency. Bearings comprise the major elements that determine the mechanical efficiency of a pump. They consist of journal sliding and bearings. Additional losses that might be categorized as mechanical losses include oil pumping and windage, which together amount to 1 to 2 Watts.

Journal bearings are sleeve types and consist of main journal bearings on the crankshaft for both types of pumps, connecting rod/journal and piston/journal bearings for reciprocating and rotary pumps respectively and wrist pin/piston/connecting rod bearings for reciprocating pumps. As a general rule, to minimize manufacturing costs, refrigeration compressors have been designed as "members" of a "family" of compressors covering a reasonably wide range of capacities, with as many common parts as practical. Common parts include motor rotor and stator laminations, cylinder housing castings, crankshafts, and connecting rod. A design that is mechanically optimized for the loads involved in the larger capacity models of the family will have larger than optimum bearings, for example, in the smaller capacity models, and, therefore, somewhat lower mechanical efficiency.

Sliding surface friction (with a lubricating oil film between parts) occurs between the piston and cylinder of both reciprocating and rotary compressors. The sliding bearing surface friction between the closely fitted cylinder and piston in a rotary pump is quite high and accounts for the difference between the mechanical efficiencies of the two types of pumps.

3.4.2.2 Pressure Losses. Low side pressure losses not only decrease the efficiency of the compressor, but reduce the gas density in the cylinder at the end of the suction stroke, reducing volumetric efficiency and cooling capacity. Major restrictions in the inlet gas flow path of reciprocating compressors include the suction muffler and the suction valve port/reed. The inlet to a rotary compressor is virtually unrestricted. The net effect is to cause a slightly higher compression ratio than the external suction and discharge conditions dictate.

Analogous pressure losses apply to the discharge side of the compressor, and result in some overcompression.

The reciprocating pump has both a suction and discharge valve plate and smaller ports while the rotary pump has only a discharge valve plate and large ports. These distinctions account for some of the relative differences in efficiencies. Increasing port sizes and valve plate sizes would benefit reciprocating pumps marginally. The value of increased discharge valve port area in terms of reduced discharge pressure loss must be balanced against the increased clearance volume of the discharge port. Rotary pumps, however, have little room for improvement in this respect.

3.4.2.3 Volumetric Efficiency. Volumetric efficiency is the ratio of the actual mass flow rate compressed by the compressor to the ideal mass flow rate based on the displacement and RPM and the suction gas density at the inlet to the compressor shell. The major losses to volumetric efficiency are

- Pressure losses between the inlet to the shell and the compressor cylinder (see brief discussion above),
- Suction gas superheating (see below). The combined effect of pressure losses and suction gas superheating is to reduce the density of the refrigerant vapor in the cylinder when the piston is at bottom dead center at the end of the suction stroke, and
- Clearance gas reexpansion.

The clearance volume is comprised of a finite space between the piston and cylinder at the end of the stroke and the discharge valve port. The piston clearance is necessary to allow for the build up of manufacturing tolerances of the piston, cylinder housing, crankshaft, and connecting rod as well as differential thermal expansion and insures against the piston mechanically striking the cylinder head and the suction valve in reciprocating pumps. The rotary pump does not have a clearance volume between the piston and cylinder and does not have a suction valve. The only clearance volume is a discharge port and the radius on the end of the vane, on the discharge port side of the vane.

The clearance volume retains a portion of the compressed gas which is not displaced from the cylinder. This retained gas re-expands on the suction stroke and occupies space in the cylinder preventing the new charge of gas being drawn in from fully occupying the cylinder. The effect of this phenomenon is to reduce the effective capacity of the pump, a corollary of which is that the pump must have a larger displacement than if its volumetric efficiency were 100%.

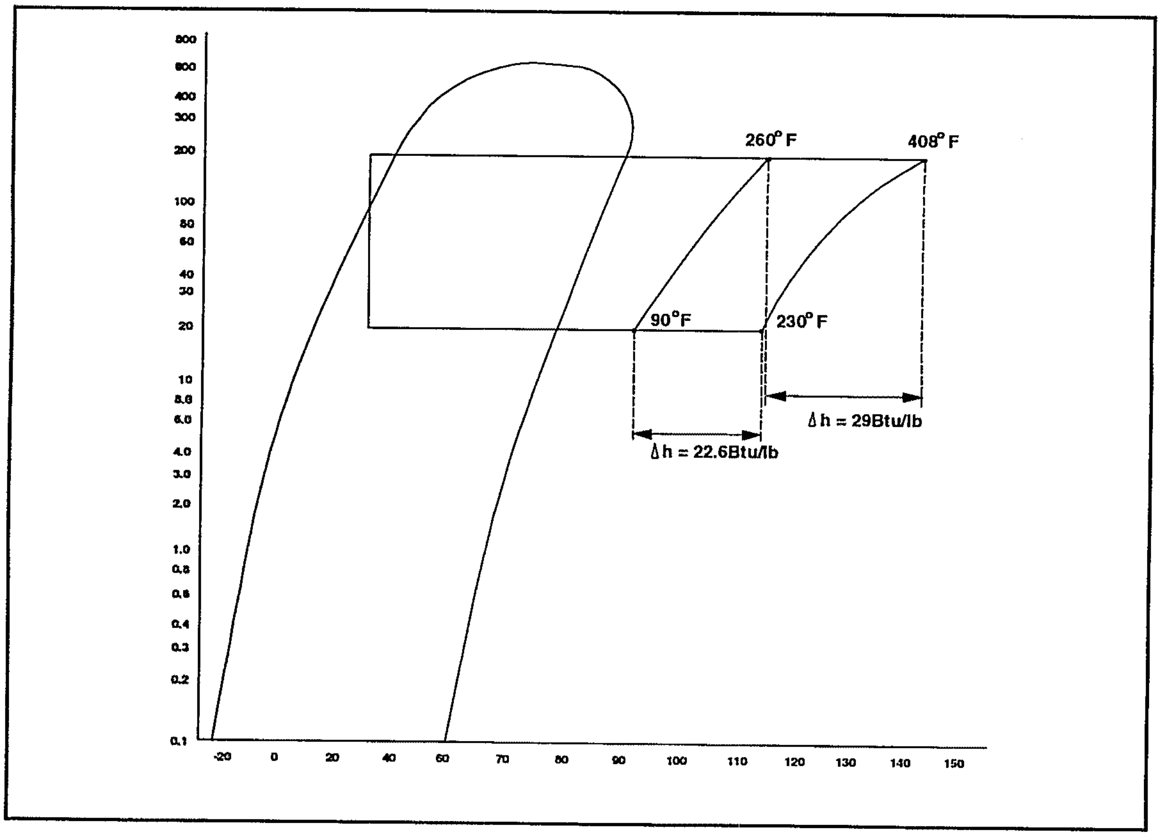
Volumetric efficiencies of reciprocating pumps tend to be in the 60% range, consistent with the not insignificant magnitude in a reciprocating compressor of all three of the above listed basic volumetric efficiency loss mechanisms. In contrast, the volumetric efficiency of rotary compressors is typically over 90%, because the design minimizes the basic losses, i.e., the direct suction minimizes both the heating of the suction gas and low side pressure losses and, as noted above, the clearance volume is minimal.

In principle, the volumetric efficiency has an effect only on pump capacity in that the compression and re-expansion of the clearance gas is a reversible process. In reality, the clearance gas compression and reexpansion processes are not reversible, primarily because of cyclic heat transfer between the gas and the cylinder walls and head and piston (the reexpanding clearance gas is somewhat cooler than during the compression process, and returns less work to the piston than was originally required to compress the clearance gas). Further, there is an indirect effect on work due to the resulting increased displacement which has a greater absolute friction loss than a smaller one. In experimental work where clearance volumes were reduced from nominal values to the minimum value possible with a given set of parts, the power input increased (because less work was returned to the piston from reexpansion of the clearance gas), but the refrigerant mass flow rate increased more in percentage terms, giving an overall improvement in compressor EER.

For example, experimental work by Westinghouse reported in 1981 showed that a 760 Btu/hr compressor with a ratio of suction volume to clearance volume of 83 and an EER of 3.4 when equipped with a modified piston, had a 9% efficiency improvement. The suction volume to clearance volume ratio was increased to 187, the pump capacity increased to 924 Btu/hr and the EER improved to 3.7.

3.4.2.4 Suction Gas Superheat. To the extent that the refrigerant gas entering the pump cylinder is above the temperature of the gas at the inlet to the compressor shell, the work of compression will increase. This is shown, in somewhat oversimplified fashion, by the pressure enthalpy diagram (Figure 3-7) which shows that the compression work required to pump a gas through a given pressure difference increases with the suction gas temperature in the cylinder at the beginning of compression. (Note that both isentropic compression curves overstate the actual discharge temperatures that would occur in a compressor having the indicated cylinder gas temperature, but is generally illustrative of the effect on compression work of increased suction gas superheat.)

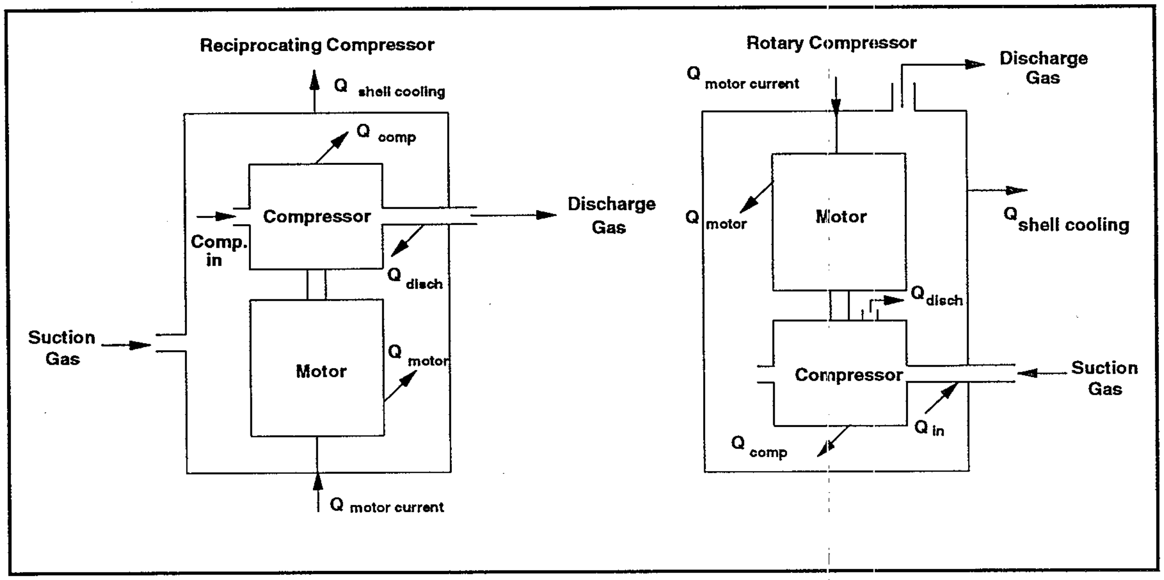
Figure 3-7: R12 Pressure/Enthalpy Diagram Effect of Superheat -- 90°F versus 230°F (-10°F Saturated Suction, 130°F Saturated Discharge, Isentropic Compression)



Some superheat is desirable to prevent wet compression and for certain other thermodynamic considerations; however, the suction gas is usually considerably superheated as a result of ambient heating and subcooling of the liquid refrigerant for increased refrigeration capacity. Figure 3-8 schematically shows the distribution of the sources of suction gas superheat for reciprocating and rotary compressors.

In reciprocating compressors, the suction gas gains heat from a number of identifiable sources. In a typical compressor, the suction gas enters the shell and mixes with gas that has been heated by the motor and other higher temperature internal surfaces, primarily the cylinder head and cylinder body and the discharge line. Next, the gas passes through the suction muffler, which is often a chamber or pair of chambers bored into the (high operating temperature) cylinder housing. Then the gas passes into the suction

Figure 3-8: Compressor Heat Distribution



manifold in the cylinder head, immediately adjacent to the high temperature discharge manifold. Finally the gas passes into the cylinder, whose walls have been heated to a temperature intermediate to the inlet and discharge gas temperatures. As a result of these multiple heating steps, the suction gas superheat may be as high as 150°F over the gas temperature at the shell connection. The suction line can be directly connected to the pump through the suction muffler provided that some type of leakage path permits the pressure in the line to equalize with the shell gas. Figure 3-9 is an example of how this could be accomplished. This is being done in some larger R/F compressors. Matsushita reported a 6 to 10% improvement in compressor efficiency for a 25°F reduction in superheat at the pump port by using this approach.

Figure 3-10 shows the theoretical increase of compressor work versus superheat over the gas inlet temperature to the shell, for the case of a 90°F gas temperature to the compressor shell. If cylinder gas temperatures were reduced from their typical 200°F+ level to the inlet temperature at the compressor shell, efficiency would be improved by 20 to 25 percent. In practice, this is not achievable, because even after a direct suction connection through an insulated muffler is implemented, significant heat transfer paths remain in the cylinder head and cylinder wall areas. The 5% to 10% efficiency improvement described in the preceding paragraph represents a practical level of improvement.

In reciprocating compressors, as suction gas superheating is reduced, more heat must be rejected through the shell or through separate cooling of the oil. To reduce suction gas superheat further and lower pump temperatures, semi-hermetic designs could be utilized. In semi-hermetic and open drive compressors, pump heat is rejected through

Figure 3-9: Low Side Compressor

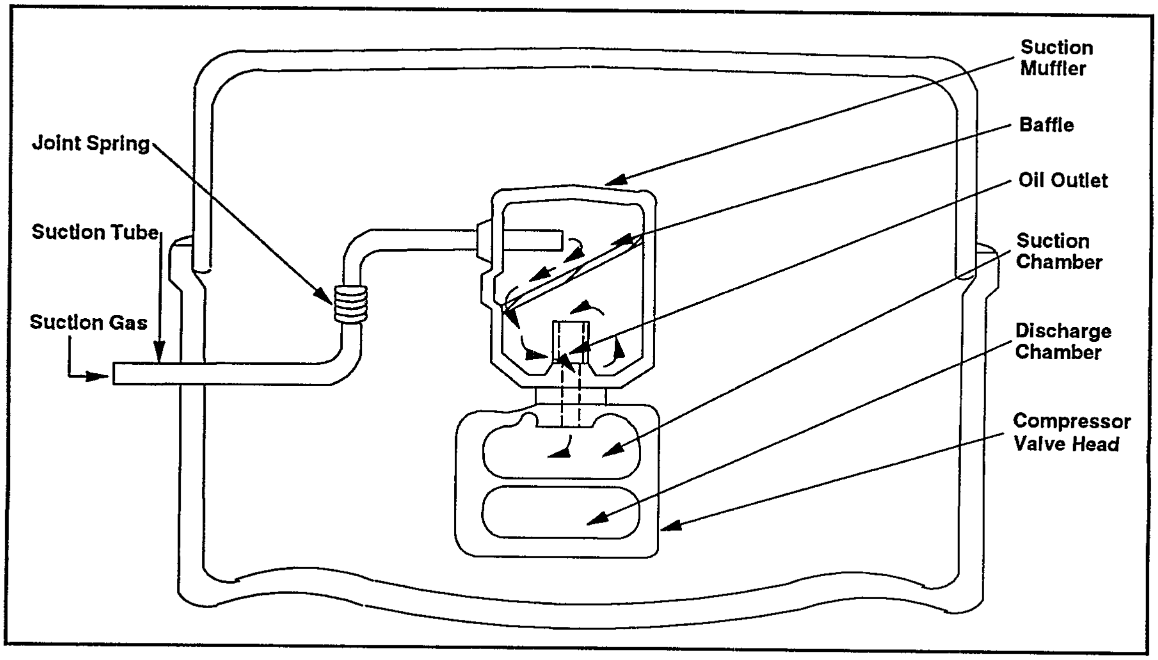
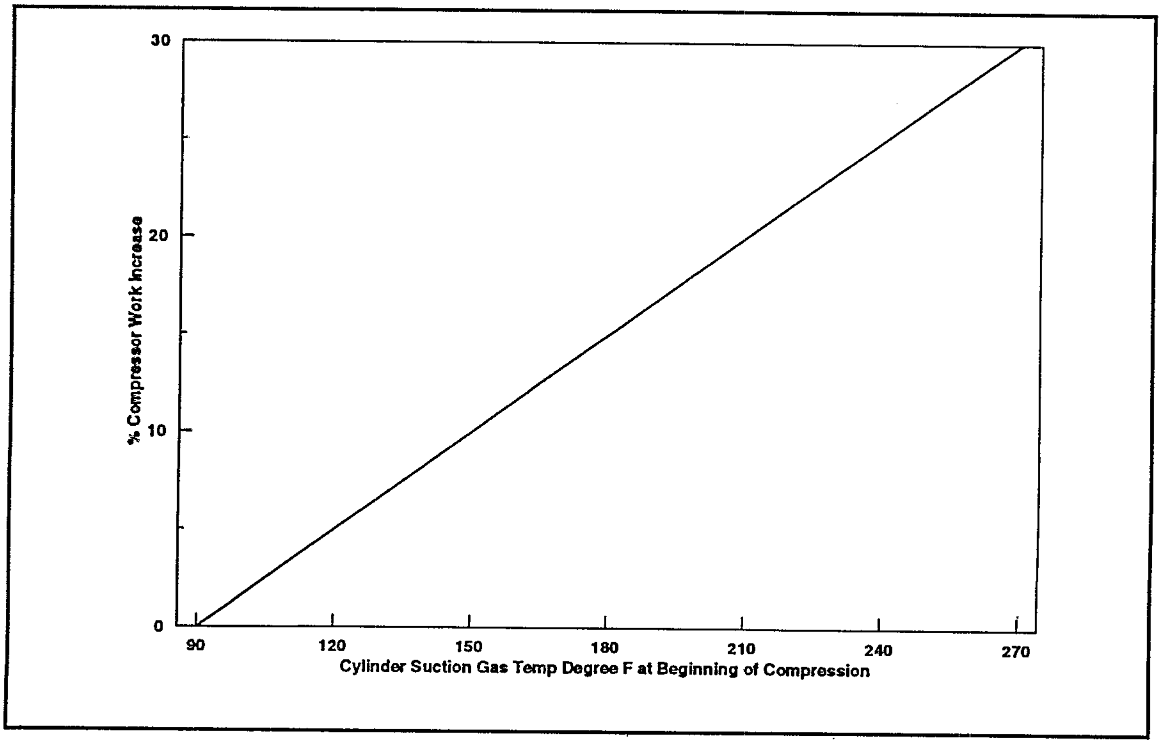


Figure 3-10: Effect of Suction Gas Temperature on Compressor Work for Constant Mechanical & Cylinder Efficiency

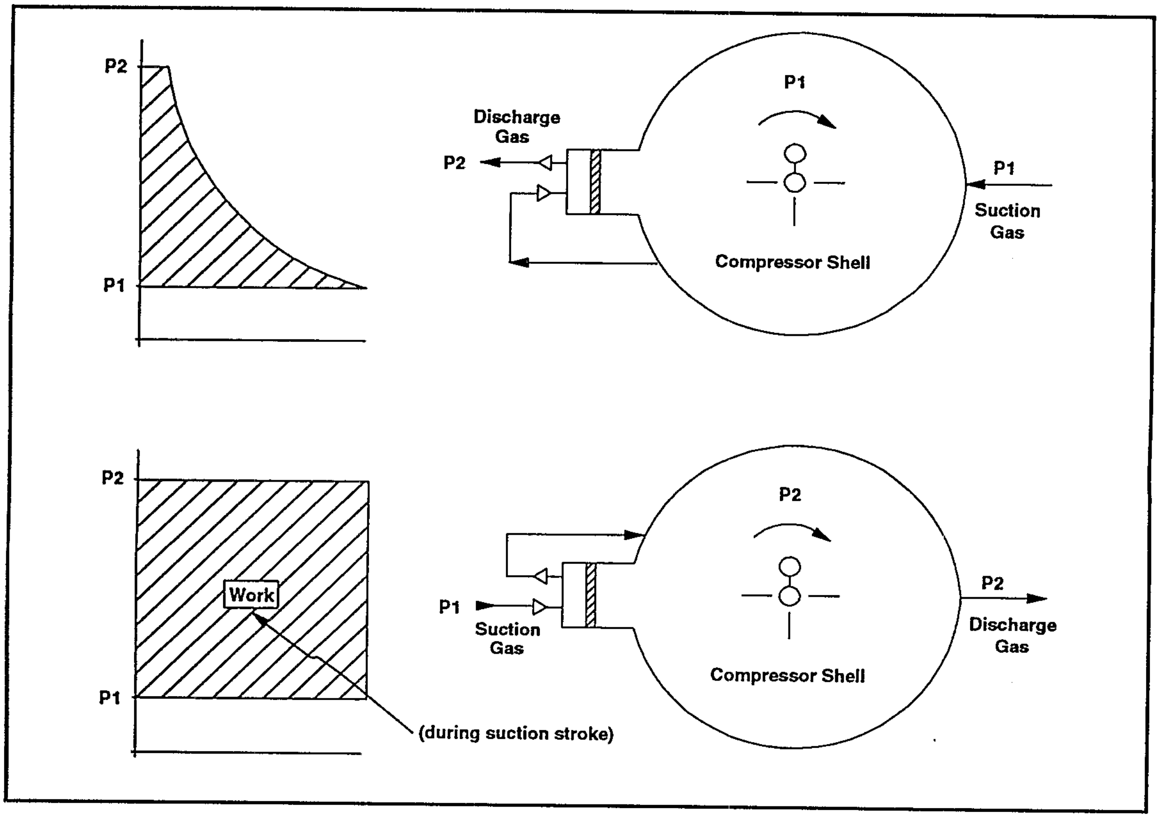


the pump housing directly to the ambient air rather than through a gas interface to a shell and then to ambient air. The net result would be lower cylinder temperatures, further reducing cylinder gas temperatures. This would require an external suction muffler and would forego the sound isolation provided by the encapsulating shell, as well as hermetic sealing of the refrigerant charge in the system.

In rotary compressors, the suction gas passes from the compressor shell directly to the cylinder inlet through a short (approximately 1 inch long) straight port in the cylinder body (see Figure 3-4). Even with heat transfer to the cylinder and rolling piston surfaces, suction gas superheat is very moderate in comparison with typical small welded hermetic reciprocating compressors, usually on the order of 40° to 50°F. This low level of superheat coupled with the negligible suction port pressure loss accounts for the high volumetric efficiency that is typical of rotaries. This level of superheating is so low that there is not much potential for additional improvement.

Unlike a rotary compressor whose crankshaft and motor operate at high side pressure, with the suction gas drawn directly into the pump, minimizing suction gas superheating, in a single cylinder reciprocating compressor the crankcase or underside of the piston operates at the suction pressure rather than the discharge pressure. The mechanical effect of operating a single cylinder compressor crankcase/piston underside at high side pressure (to allow direct suction to the compressor cylinder) is to significantly increase the amount of work performed by the piston on each stroke (the theoretical network per complete crankshaft resolution is the same in either case), significantly increasing bearing loads and losses. As illustrated in Figure 3-11, with a low side crankcase, the theoretical pressure volume process follows the upper PV diagram. During the suction stroke, no pressure differential (neglecting losses) acts across the piston, and during the discharge stroke all of the PV work is performed. With a high side crankcase, as shown in the lower diagram, the refrigerant vapor in the cylinder on the top side of the piston still would follow the upper PV process. However, during the suction stroke, the differential between high and low side pressure would act across the piston. The resulting work done by the piston would be the area indicated in the lower PV diagram, more than double the basic gas compression work. While this work would be "returned" to the piston during the discharge stroke, the work on each individual stroke would be handled mechanically; during the suction stroke, bearing loads, and losses would nearly be doubled.

Figure 3-11: Low Side versus High Side Crankcase in a Single Cylinder Compressor



4.0 Options for Improvement and Their Cost: Large Compressors

4.1 Reciprocating Compressors

The "large" reciprocating compressors currently used in the majority of U.S. manufactured domestic R/F have nominal capacities ranging between 600 Btu/hr and 1300 Btu/hr. In this capacity range, the maximum efficiency level of production compressors has increased significantly over the past decade, as indicated in Figure 4-1. In 1980, the maximum compressor EER in this capacity range was approximately 4.0 Btu/Watt-hr. By 1992, available EERs had increased from 5.3 to 5.5. Currently, in early 1993, limited numbers of compressor samples having an EER of 6.0 (using a high efficiency, electronically commutated permanent magnet DC motor) are available.

Figure 4-1: Improvement in Efficiency of "Large" Compressors for Domestic Refrigerator/Freezers over the Past Decade (EER of Isentropic Compressor with 100% efficient motor is 9.3 Btu/Watt-hr)

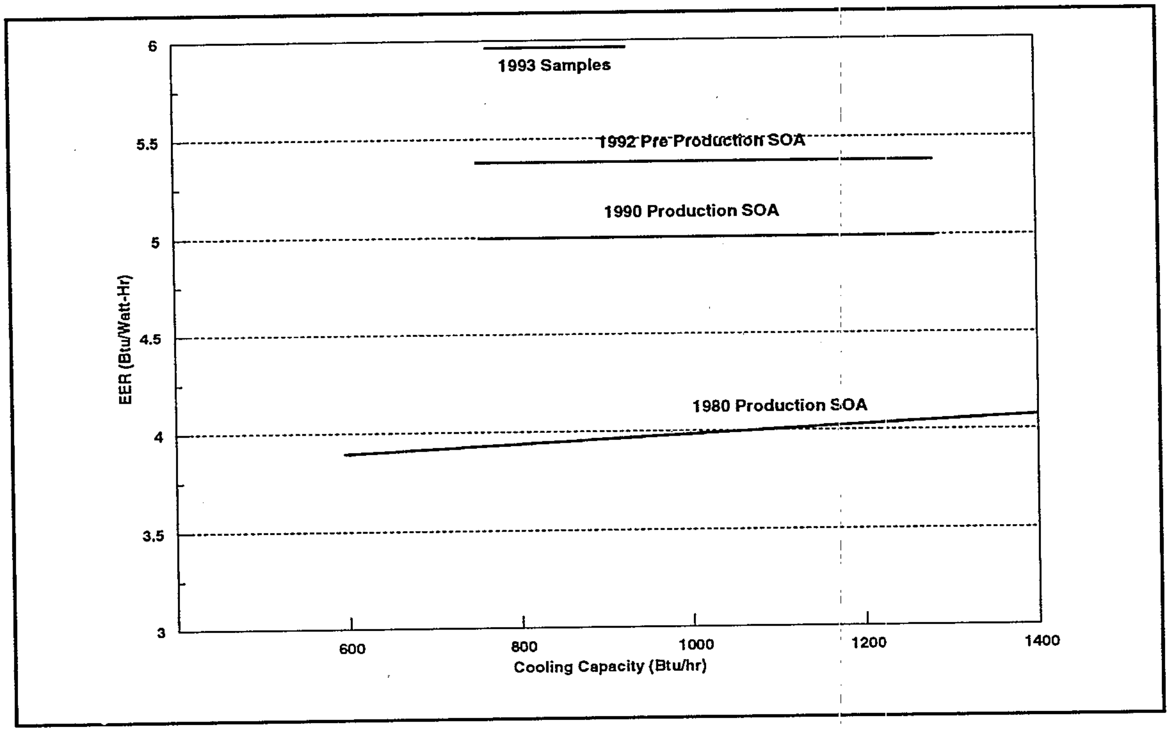


Table 4-1 is a breakdown of the input power for large compressors at the four efficiency levels shown in Figure 4-1. Significant reductions in motor losses and suction gas heating losses provided most of the performance improvement between the 4.0 EER level of 1980 and the 5.0 EER level of the best current production compressors. Further improvement to the 5.3 EER level has come from incremental reductions of mechanical losses (primarily through the use of a reduced viscosity lubricant), motor losses, and suction gas heating. The OEM cost premium for this level of compressor efficiency is on the order of $5. EER's on the order of 5.5 to 6.0 have been obtained through further incremental loss reduction, as indicated in the last column of Table 4-1. The design measures include using the highest possible efficiency electric motor (electronically commuted permanent magnet rotor DC motor) and reduced suction gas heating losses (probably requiring active cooling of the cylinder body and head, as shown in Figure 3-6). This level is probably approaching the practical limits; the necessary design measures would add about $15 to the OEM price of the compressor.

Table 4-1: "Large" Reciprocating Compressor Input Power Distribution vs. Efficiency Level

	1980 Production State of the Art (EER = 4.0)	Current Production (EER = 5.0)	Current Production State of the Art Samples (EER = 5.3)	Future Goal (EER = 6.5)
Input Electric Power	100%	100%	100%	100%
Motor Loss	23%	17%	16%	8%
Mechanical Losses	9%	8%	7%	6%
Suction Gas Heating	15%	12%	11%	8%
Discharge Port Losses	2%	2%	2%	2%
Low Side Pressure Losses	4%	4%	4%	3%
Compression and Expansion Losses (clearance gas expansion)	3%	2%	2%	2%
Piston Blow By/Internal Leakage	1%	1%	1%	1%
Power Delivered to Gas Compression	43%	54%	57%	70%

* Based on reversible adiabatic compression of actual delivered mass flow rate from shell inlet conditions to discharge pressure.

4.1.1 Non-Lubricated Linear Free Piston Reciprocating Compressor

In a project sponsored by the U.S. EPA, Sunpower has developed a prototype, non-lubricated linear free piston compressor. The piston is driven by an electronically-driven linear, permanent magnet motor. A measured EER at standard conditions of 6.2 has been reported (References 40, 42). Efficiency improvements are attributed to high (94%) motor efficiency, having only minimal friction losses in the piston/cylinder gas bearing, offset to some extent by additional piston blow by and some additional valve losses, both due to absence of the sealing effect of the oil film, piston spring losses, and additional losses due to the unbalanced reciprocating motion of the piston. The claimed potential for further efficiency improvement has not yet been demonstrated. Significant issues include the level of reliability and operating life (40,000 to 80,000 hours without failure in well over 90% of units is routine in current compressors) that can be attained with an unlubricated free piston device and the cost to manufacture, including necessary high energy permanent magnet materials and electronic drive in the motor.

4.2 Rotary Compressors

In the "large" compressor capacity range, current production rotary compressors are approximately competitive with current production reciprocating compressors with respect to both cost and efficiency; within the range of their own manufacturing processes and cost/performance/reliability trade offs, different manufacturers have placed greater or lesser emphasis on each technology.

Table 4-2 summarizes the breakdown of losses for current production and future, improved efficiency "large" rotary compressors. The efficiency level of the middle column was reported in Reference (38), and was the result of an intensive effort to reduce all of the major losses (except the motor losses) to the minimum possible level. A significant reduction in blow-by loss was achieved by reducing the rolling piston to cylinder clearances by a factor of approximately two. Incremental reduction in clearance gas re-expansion loss was obtained through reduction of the discharge port volume and an incremental reduction of mechanical losses was achieved through optimization of bearing diameters, lengths, and clearances. To date, no rotary compressor is in production at this EER level, perhaps being indicative of the difficulty in mass production of holding the tolerances required to allow the reduced clearances needed to minimize blow by losses.

Table 4-2: "Large" Rotary Compressor Input Power Distribution vs. Efficiency Level

	Typical Rotary (EER = 4.7)	Reference (38) (EER = 5.1)	With Best Motor (EER = 5.6)
Input Electric Power	100%	100%	100%
Motor Loss	22%	22%	16%
Mechanical Losses	10%	8%	9%
Suction Gas Heating	3%	3%	3%
Discharge Port Losses	4%	3 1/2%	3 1/2%
Low Side Pressure Losses	2%	2%	2%
Compression and Expansion Losses	3%	2 1/2%	2 1/2%
Piston Blow By/Internal Leakage	5%	4%	4%
Power to Compressor - Gas Delivered*	51%	55%	60%

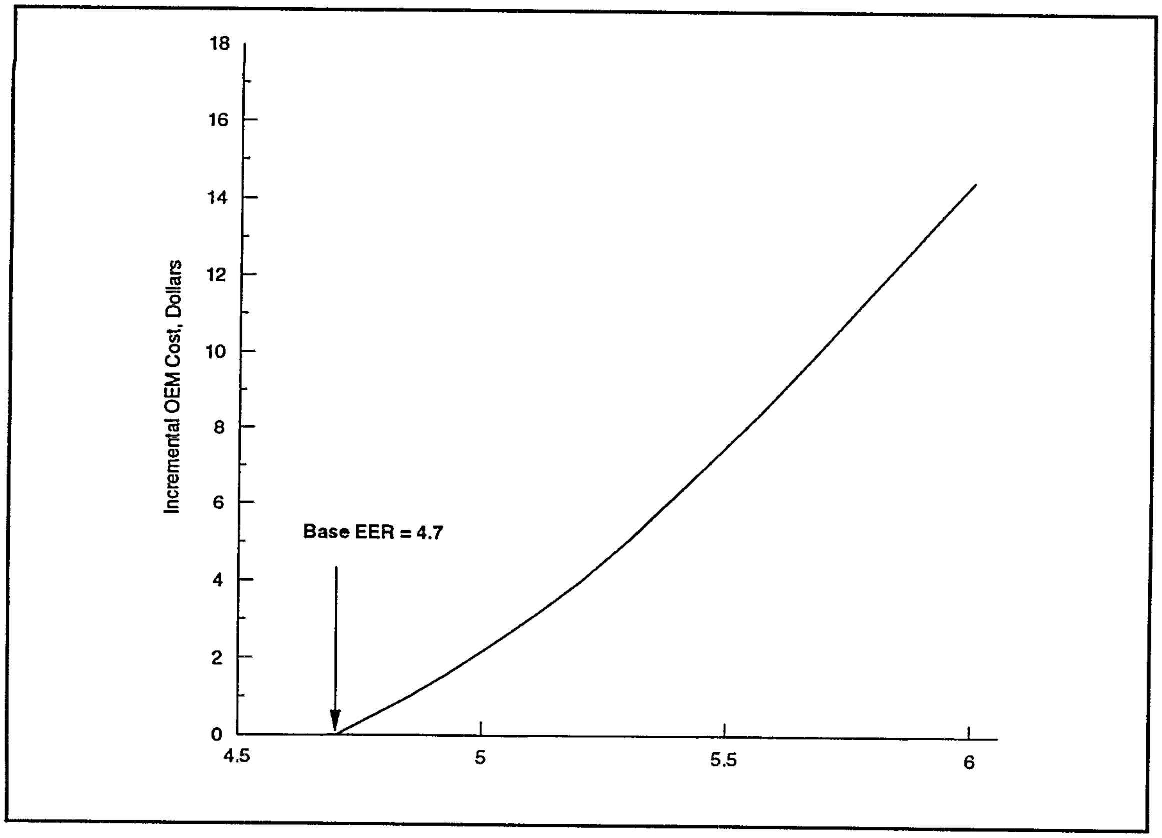
* Based on reversible adiabatic compression of actual delivered mass flow rate from shell inlet conditions to discharge pressure.

The last column shows the efficiency level resulting from substituting a high efficiency motor for the motor used in the Reference (38) compressor. Due to the higher operating temperature and smaller diameter of the motor in a rotary compressor, the best motor efficiency is approximately 2 percentage points less than that of the motor for a comparable capacity reciprocating compressor. The resulting EER of 5.6 is essentially comparable to the estimated limiting efficiency range for a "large" reciprocating compressor.

4.3 Cost of Increased Efficiency

As discussed in Section 3.3, manufacturers are generally reluctant to discuss their OEM pricing policies in any detail, citing both domestic and foreign competition and confidential relationships with their customers. Figure 4-2 plots the estimated OEM cost premium of increased efficiency for large compressors. The plot is based on numerous informal discussions with compressor manufacturers and other industry participants. A significant portion of the cost increase is attributable to incremental increases in motor costs, as the maximum feasible efficiency of the motor is approached.

Figure 4-2: Estimated Incremental Cost of Improved Efficiency Large Compressors



No attempt has been made to estimate the magnitude of the capital investments in R&D and new tooling that would be required to develop new, higher efficiency compressors.

5.0 Options for Improvement and Their Cost: Small Compressors

As indicated in the preceding sections, the efficiencies of the smaller compressors that are the focus of this study and will be needed to effectively utilize improved cabinet thermal design and dual evaporator cycles currently lag the efficiency that is available in the best compressors utilized in larger refrigerators and freezers. Where the efficiency level of larger compressors has been improved substantially since 1980, only limited improvements in efficiency have been implemented in the smallest compressors. In order to achieve EERs in the 200 to 600 Btu capacity range comparable to those currently available in the higher capacity machines efficiency improvements in both the motor and pump are required.

In this section, the potential for improvements is discussed in terms of specific options for improving small reciprocating compressors and rotary compressors.

The emphasis is on the 200 to 400 Btu/hr capacity range needed for dual loop systems. Owing to the lack of current production, high efficiency, small compressors, a set of loss scaling relationships have been developed to assess the extent to which the efficiency improvements obtained in larger compressors might be obtained in small compressors through similar design modifications.

5.1 Reciprocating Compressors

Table 5-1 is an estimate of the maximum efficiency level that is attainable in a small, 200 Btu/hr nominal capacity reciprocating compressor, based on applying the loss scaling factors discussed in Section 5.4, to the maximum feasible efficiency large compressor (nominal capacity 800 Btu/hr, EER 6.0) discussed in Section 4.1. The resulting EER of 5.3 represents both a substantial improvement over current production small compressors and a significant R&D challenge.

Consistent with the upper limits on small compressor performance estimated in the preceding paragraph, Table 5-2 is a breakdown of the input power in a nominal 200 Btu/hr capacity compressor at current low efficiency levels, at an intermediate efficiency level reached primarily through improved motor performance, and at a maximum feasible efficiency level, where measures have been taken to reduce all losses. The intermediate efficiency level is reached through motor efficiency improvement. The maximum feasible efficiency level is achieved by using the highest efficiency motor and implementing measures to reduce mechanical losses, clearance volume and suction gas heating.

Table 5-1: Estimated Small Reciprocating Compressor Limiting Efficiency and Input Power Distribution, Based on Loss Scaling

	Large Reciprocating at Limiting Efficiency (EER = 6.5)	Loss Scaling Factor**	Loss Scaling Factor @ 1/4 Capacity***	Limiting Efficiency @ 200 Btu/hr
Input Electric Power	100%	N/A	N/A	100%
Motor Loss	8%	Table 3 - 5	+2%	10%
Mechanical Losses	6%	Capacity $^{-1/6}$	1.26	7.5%
Suction Gas Heating	8%	Capacity $^{-1/6}$	1.26	10%
Discharge Port Losses	2%	--	--	2%
Low Side Pressure Losses	3%	--	--	3%
Compression and Expansion Losses	2%	Capacity $^{-1/3}$	1.6	3.2%
Piston Blow By/Internal Leakage	1%	--	--	1%
Power Delivered to Gas Compression*	70%	N/A	N/A	63%
			EER	5.7

* Based on reversible adiabatic compression of actual delivered mass flow rate from shell inlet conditions to discharge pressure.

** Scaling factors are discussed in 5.4

*** Comparison of 200 Btu/hr to 800 Btu/hr nominal capacities

Table 5-2: Small (200 Btu/hr Nominal Capacity) Reciprocating Compressor Input Power Distribution at Several Efficiency Levels

	Low Efficiency, Current Production SOA (EER = 3.2)	Intermediate Efficiency (EER = 4.2)	Maximum Efficiency EER = 5.7
Input Electric Power	100%	100%	100%
Motor Loss	28%	18%	10%
Mechanical Losses	10%	10%	8%
Suction Gas Heating	16%	16%	10%
Discharge Port Losses	2%	2%	2%
Low Side Pressure Losses	4%	4%	3%
Compression and Expansion Losses	4%	4%	3%
Piston Blow By/Internal Leakage	1%	1%	1%
Power Delivered to Gas Compression*	35%	45%	63%

* Based on reversible adiabatic compression of actual delivered mass flow rate from shell inlet conditions to discharge pressure.

5.2 Rotary Compressors

Table 5-3 is an estimate of the maximum efficiency level that is attainable in a small, 200 Btu/hr nominal capacity rotary compressor, based on applying the loss scaling factors discussed in Section 5.4, to the maximum feasible efficiency large compressor (nominal capacity 800 Btu/hr, EER 5.6) discussed in Section 4.2. The resulting EER of 4.4, which is considerably less than the estimated limiting efficiency for small reciprocating, is indicative of the extent to which scaling factors limit the efficiency potential of rotary compressors for the smallest capacity applications.

Table 5-3: Estimated Small Rotary Compressor Limiting Efficiency and Input Power Distribution, Based on Loss Scaling

	"Large" Rotary at Limiting Efficiency (800 Btu/hr, EER = 5.6)	Scaling Factor**	Loss Scaling Factor @ 1/4 Capacity***	Small Rotary Limiting Efficiency @ 200 Btu/hr
Input Electric Power	100%	N/A	N/A	100%
Motor Loss	16%	Table 3 - 5	+2%	18%
Mechanical Losses	9%	Capacity $^{-1/6}$	1.26	12%
Suction Gas Heating	3%	Capacity $^{-1/3}$	1.6	5%
Discharge Port Losses	3 1/2%	--	--	3 1/2%
Low Side Pressure Losses	2%	--	--	2%
Compression and Expansion Losses	2 1/2%	--	--	2 1/2%
Piston Blow By/Internal Leakage	4%	Capacity $^{-2/3}$	2.5	10%
Power Delivered to Gas Compression*	60%	N/A	N/A	47%
			EER	4.4

* Based on reversible adiabatic compression of actual delivered mass flow rate from shell inlet conditions to discharge pressure.

** Scaling factors are discussed in 5.4

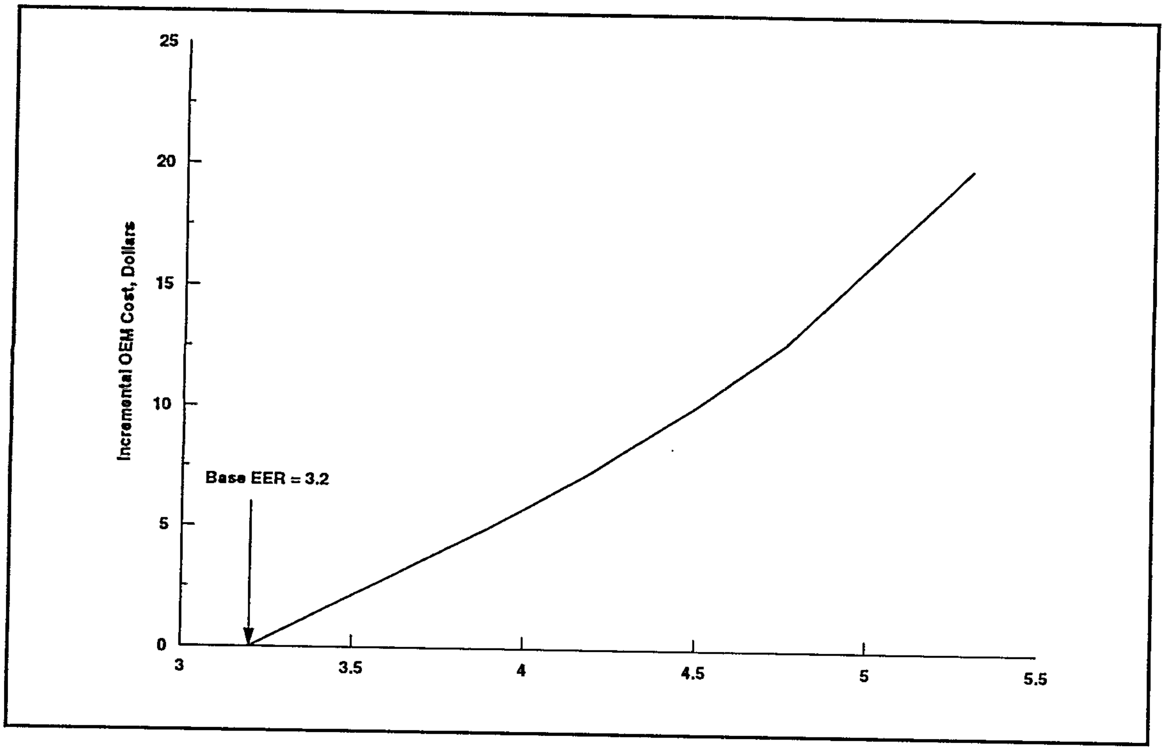
*** Comparison of 200 Btu/hr to 800 Btu/hr nominal capacities

5.3 Estimated Cost of Improved Efficiency Small Compressors

As discussed in Section 3.3, manufacturers are generally reluctant to discuss their OEM pricing policies in any detail, citing both domestic and foreign competition and confidential relationships with their customers. Figure 5-1 plots the estimated OEM cost premium of increased efficiency for small reciprocating compressors. The plot is based on numerous informal discussions with compressor manufacturers and other industry participants. A significant portion of the cost increase is attributable to incremental increases in motor costs, as the maximum feasible efficiency of the motor is approached.

No attempt has been made to estimate the magnitude of the capital investments in R&D and new tooling that would be required to develop new, higher efficiency compressors.

Figure 5-1: **Estimated Incremental Cost of Improved Efficiency Small Compressors**



5.4 Approach to Scaling of Losses

To estimate the potential efficiency level that could be obtained in small compressors through the application of the same design features that have been used to improve large compressor efficiencies, a set of simple scaling relationships relating the magnitude of each loss to the capacity of the compressor was developed, for both reciprocating and rotary compressors. Table 5-4 summarizes both sets of loss scaling relationships, giving the variation with capacity of the loss as a percentage of the total power input. The assumptions and rationale for the treatment of each loss are discussed briefly below. These scaling relationships generally apply to the 200 Btu/hr to 800 Btu/hr nominal capacity range, at a constant 2-pole motor speed of approximately 3500 RPM. A basic assumption underlying these scaling relationships is that the small capacity design is geometrically similar to the large capacity design, i.e. all dimensions are scaled by the same factor, with the exception of close clearances. To a first order, the capacity and input power are proportional to the displacement, which is proportional to the cube of the linear dimensions, e.g. diameter, length. This simplified treatment ignores many complexities and interrelations of important design variables, such as the temperature dependence of oil viscosity; as such it can be regarded in the aggregate as being generally indicative of the magnitude of the effect of compressor capacity on the losses and the resulting efficiency potential.

Table 5-4: **Scaling of Losses in Reciprocating and Rotary Compressor**

	Scaling Rule for Loss vs. Capacity	
	Reciprocating	**Rotary**
Motor Loss	Small - Table 3-5	Small Table 3-5
Mechanical Losses	Capacity $^{-1/6}$	Capacity $^{-1/6}$
Suction Gas Heating	Capacity$^{-1/6}$	Capacity $^{-1/3}$
Discharge Port Losses	No Scale Effect	No Scale Effect
Low Side Pressure Losses	No Scale Effect	No Scale Effect
Clearance Gas Reexpansion	Capacity $^{-1/3}$	No Scale Effect
Piston Blow By/Internal Leakage	No Scale Effect	Capacity $^{-2/3}$

Motor Losses. The treatment in Section 3.4.1 is the basis for relating losses to capacity, with Table 3-5 summarizing the relationship between capacity and motor efficiency.

The maximum motor efficiency for a rotary compressor is about 2 percentage points less than the values indicated in Table 3-5, because of the higher operating temperature and larger aspect ratio of the motor.

High and Low Side Pressure Losses. For both recips and rotaries, the effect of geometrically scaling, at constant rotational speed, vapor passages and valve port diameters is to reduce gas velocities and pressure losses, complicated by considerations such as the effect of inlet and discharge valve reed stiffness on pressure drops. The overall effect of scale is probably small, and is neglected in this treatment. This is the one area where the scaling factors are favorable to small capacities.

5.4.1 Reciprocating Compressors

Mechanical Losses. Mechanical losses include shaft bearing friction losses and piston-cylinder wall sliding friction (hydrodynamic bearing oil film shear stress in both cases). Other, smaller mechanical losses include oil pumping and windage.

- Piston friction power dissipation is the product of the average oil film shear force acting on the piston (oil viscosity x piston sliding surface area x average velocity ÷ clearance) and the average piston velocity. The basic scaling relationships are:
  - Average velocity $\propto$ diameter
  - Piston sliding surface area $\propto$ diameter squared
  - Oil viscosity is constant
  - The diametral piston to wall clearance is constant, on the order of 0.0001 to 0.0002 inches, dictated by the minimum practical level for selective assembly
  - Overall, the power dissipation is proportional to d^4, the ratio of power dissipated to input power is $d^4/d^3 = d =$ capacity$^{1/3}$

- Shaft bearing friction: for a given journal bearing lubricant viscosity, L/D ratio, clearance to diameter ratio, and operating eccentricity, at constant rotational speed the power dissipated in the bearing is proportional to the design load, which varies with the cube of the shaft diameter. The load, which originates primarily from gas pressure acting on the piston is proportional to the piston diameter squared. The ratio of the power loss (proportional to the bearing loads) to the total power input is $d^2/d^3 = d^{-1} = \text{capacity}^{-1/3}$.

- The net effect of the relatively small piston friction plus the bearing friction is estimated to be: mechanical power losses/total input power = $\text{capacity}^{-1/6}$.

Suction Gas Heating. As discussed below, under rotaries, suction gas heating varies with surface area to volume ratio, given a constant internal temperature distribution. In small recips, suction gas heating is a more significant loss, and affects the temperature distribution. With forced air cooling of the compressor shell, average internal temperatures will be reduced to a greater extent at smaller capacities. Overall, a lesser dependence of suction gas heating on scale is assumed: $\text{capacity}^{-1/6}$.

Clearance Gas Reexpansion. The magnitude of the ratio of this loss to total input power depends on the ratio of the clearance volume to the displacement. In larger, high efficiency compressors, this ratio has been reduced to the minimum practical level, and involves the handling of piston, cylinder, and connecting rod length tolerances, with a resulting minimum piston crown to cylinder head clearance at top dead center. The minimum tolerance based clearance will not scale down with reduced dimensions to any significant extent. The piston stroke does scale, so the ratio of clearance volume to displacement will be proportional to d^{-1} or capacity to $^{-1/3}$.

Piston Blow-by. In a recip, the clearance between the piston and cylinder is small (diametral clearance between 0.0001 and 0.0002 inch) and filled with an oil film. The loss is generally less than 1% and scaling was neglected in this treatment.

5.4.2 Rotary Compressors

Mechanical Losses. Crankshaft bearing loads and losses and piston sliding losses scale in the same fashion as with recips. (overall loss to power output ratio scales with capacity$^{-1/6}$).

Suction Gas Heating. This is a fairly small loss in a rotary, and consequently moderate changes in the loss will not have a significant effect on temperature levels throughout the compressor. The magnitude of the heat transfer from the high side shell and the cylinder walls to the suction gas will be proportional to areas, or d^2. The ratio of suction gas heating to capacity is proportional to $d^2/d^3 = d^{-1} =$ capacity$^{-1/3}$.

Clearance Gas Reexpansion. The clearance volume in a rotary will scale with the overall dimensions; therefore no significant scale effect is expected.

Piston Blow-By. This is a major loss in "large" rotaries, kept under control by rigorous control of all internal clearances to the minimum practical level. It is doubtful whether significantly smaller clearances can be maintained in smaller capacity rotaries. Based on the assumption that the clearances are constant, the blow by loss is proportional to the perimeter which is proportional to d. The ratio of the loss to the capacity is $d/d^3 = d^{-2} =$ capacity$^{-2/3}$.

6.0 Potential Value of Improvements

Table 6-1 summarizes the annual energy savings obtainable through improved compressor efficiencies. These savings have been evaluated for two basic R/F (top mounted freezer) configurations: single evaporator/single compressor and dual loop. The baseline, single evaporator configuration is representative of a configuration whose efficiency approximates the 1993 Federal Energy Efficiency Standards and includes the following basic design features:

Table 6-1: Energy Savings

	Compressor Efficiency			
	EER	Comment	Annual Energy kWh*	Annual Energy Saving** kWh
Large Compressor Single Evaporator	4.85**	Readily available	685	--
	5.0	Best production compressor	669	16
	5.5	1993 production	618	67
	6.0	Approaching practical limits	570	115
Small Compressors Dual Loop	3.6/4.7	Currently available	602	83
	4.6/5.1	With efficient motor	537	148
	5.3/5.5	Approaching practical limits	498	187

* At DOE test conditions, taken to be representative of actual use

** Annual energy savings relative to large compressor/single evaporator baseline

- Cabinet volumes (cubic feet):
  - freezer 4.6
  - fresh food 13.4
  - total 18.0
  - adjusted volume 20.9

- Federal energy efficiency standard:
  - 1990 962 kWh/yr
  - 1993 690 kWh/yr

- 4.85 EER, 893 Btu/hr compressor (readily available, moderate cost, reasonable efficiency, provides typical pulldown capacity).

- Cabinet insulation is polyurethane foam with R=8°F/in per Btu/hr-ft^2
  - average freezer wall thickness: 2 3/8 inch
  - average fresh food compartment wall thickness: 1 3/4 inch
  - insulation thickness in doors: 1 5/8 inch

- Standard evaporator (2 rows deep x 8 rows high)

- Efficient (PSC type) fans:
  - evaporator 50 CFM, 6.8 Watts
  - condenser 90 CFM, 6.8 Watts

- Electric resistance anti-sweat heaters: 5.5 Watts for mullion (the energy analysis assumes that the heater power is on one half of the time to approximate the DOE test procedure)

- Vapor-line anti-sweat heat for cabinet door flange (cycle average of 4 Watts)

- Defrost energy is assumed to be constant across all cases studied. Variations in cycle time are assumed to be compensated by adjustments to the compressor timer interval.

This baseline configuration is similar to the baseline configuration that was used in the Motor Technology Study (Reference 3) to evaluate efficiency improvements obtainable from variable speed comprerssor and fan operation. The dual loop configuration uses the same cabinet as the single evaporator configuration. The differences are:

- Evaporators - each evaporator is the same size and capacity of the single evaporator.

- Fans - each evaporator and condenser fan is the same size as in the single evaporator cycle

- Compressors, currently available small compressors:
  - freezer: 4.7 EER, nominal capacity is 440 Btu/hr
  - fresh food: 3.6 EER, nominal capacity is 208 Btu/hr

For each of these basic configurations, the annual energy consumption at the DOE test conditions was calculated for the baseline compressor efficiency, and for progressively increased compressor efficiency, as indicated in Table 6-1, consistent with the discussions in Sections 4 and 5. For both single evaporator and dual loop systems, the annual energy savings are relative to the baseline single evaporator configuration.

The value of these energy savings is tabulated in Table 6-2 assuming an electric energy cost of 8¢ per kWh, for a range of real (after inflation) discount rates between 2 and 10 percent, over the projected 15 year life of the refrigerator. The discount rates cover a range between real, after tax returns to savings accounts at the low end of the range, to after inflation credit card interest rates at the high end of the range. For real discount rates in the range that consumers would rationally choose for safe investments (at the low end of the range), the present value of the saved energy is on the order of $50 to $100 greater than the incremental cost, at retail, of the increased compressor efficiency.

Table 6-2: Present Value of Energy Savings Obtained with Increased Efficiency Compressors

	Compressor EER	Savings kWh/yr**	Present Value for Discount Rate			Cost of Improvements (in U.S. Dollars at Retail)
			2%	5%	10%	
Large Compressor Single Evaporator	4.85*	--	--	--	--	0
	5.0	16	17	14	10	5
	5.5	67	70	58	43	10
	6.0	115	120	99	73	20
Small Compressors Dual Loop	3.6/4.7	83	87	71	3	20
	4.6/5.1	148	155	127	94	40
	5.3/5.5	187	196	161	169	80

* Baseline
** Relative to baseline

7.0 Areas for R & D

As discussed in the preceding sections, considerable potential exists for improving the efficiency of the small hermetic compressors that would be used in dual loop designs. The areas for improvement include increasing motor efficiency and reducing losses in the compressor: suction gas superheat, clearance gas re-expansion, valve losses, and mechanical losses.

The techniques for designing higher efficiency induction motors are well known in the hermetic motor industry. The major issues involved from the industry viewpoint are product engineering and marketing issues of cost vs. market size, and integration with the compressor. Beyond these commercialization issues, provision of a high efficiency motor for a small compressor is relatively straightforward. Because the motor is an integral part of the compressor package, high efficiency motor development is best carried out in the context of high efficiency compressor development; integration of a high efficiency motor with an existing small compressor represents a low risk, readily implementable means of obtaining a significant increase in small compressor efficiency. Early development of small, increased efficiency compressors would provide important support to current programs to demonstrate the performance potential of R/F's with dual loop systems and/or super insulated cabinets. The focus of a project should be the small, 200 Btu/hr nominal capacity (at standard rating conditions) compressor intended for the fresh food compartment of a dual loop refrigerator/freezer of typical (18 to 22 cubic feet) refrigerated volume.

The results of the performance and payback analysis discussed in Section 6 clearly indicate that significant energy savings can be obtained with a dual loop refrigeration system in a refrigerator/freezer, if the compressor efficiencies are the levels that can be attained with the improvements discussed in Sections 4 and 5, with the energy savings at current electricity prices providing a 3 to 5 year payback of the projected $40 to $80 increase (at retail) in compressor cost. To support ongoing efforts to develop and demonstrate this performance, early development of a small, maximum efficiency compressor is justified. As discussed in Section 5, the target performance for this compressor, at standard rating conditions (-10/130/90/90/90), would be:

- 5.0 EER
- 200 Btu/hr

The basic approach would be to utilize the highest efficiency motor available and modify current small compressor designs to significantly reduce suction gas superheating. Other loss areas should also be systematically examined for improvement potential. With the continuing acceleration of the CFC phase out, a non-CFC refrigerant and compatible lubricant should be utilized.

8.0 Conclusions

This study has dealt primarily with the extent to which the efficiency of small welded hermetic refrigeration compressors used in domestic refrigerators and freezers (having nominal capacities well below 800 Btu/hr) can be increased to levels approaching the state of the art (in production) for larger capacity welded hermetic domestic refrigeration compressors.

The efficiency levels now attained in the compressors used in the majority of domestic refrigeration freezer applications (EERs between 4.8 and 5.3 Btu/Watt-hr, for nominal capacities between 700 and 1100 Btu/hr) are the result of development efforts that were initiated in the late 1970s. The hermetic refrigeration compressors in this capacity range that were available then had EER values similar to today's small compressors. Development efforts have been focussed on achieving significant efficiency gains, but within very tight cost constraints and high reliability requirements. Mass scale production and specification in R/Fs of the resulting high efficiency (but somewhat higher cost) compressors has been driven by a combination of consumer demand for higher efficiency refrigerators and the Federal energy efficiency standards.

In response to continuing tightening of the energy efficiency standards, compressor manufacturers have continued to invest in the development of higher efficiency, large compressors. Preproduction samples having an EER of 5.3 to 5.5 became available to R/F manufacturers in 1991. Production of these compressors began in 1992. In anticipation of tighter standards in the future, there are continuing R&D programs aimed at developing still higher efficiency compressors. Samples of ECM motor driven compressors having an EER of 6.0 have become available in early 1993. The practical, physical limit for the EER of compressors in this size range is approximately 6.3 to 6.5.

To date, the small capacity compressors used in small, countertop refrigerators have not been involved in the efficiency upgrade programs because there has been no market pressure to improve the efficiency of these small refrigerators that is comparable to market cost pressures (retail prices for small countertop refrigerators typically range between $99 and $199; in relationship to these retail price levels, $10 to $20 to upgrade the efficiency of the compressor is a significant increment in cost).

Efficiency in the 200 to 600 Btu range of compressors can be improved with the application of current technology. Reciprocating compressors can be scaled down to low capacity levels with much less efficiency penalty than rotaries. Efficiency levels on the order of 5.0 EER can be attained at the small end of this range through application of current technology. The major design changes needed to reach this performance level are:

- Substitution of higher efficiency PSC or RSCR motors for the RSIR motors used in current small compressors,

- Implementing design features to reduce suction gas superheating in the compressor, including direct suction through an insulated muffler and forced air cooling of the compressor shell,

- Reducing clearance volume at piston top dead center, to improve volumetric efficiency, and reduce clearance gas reexpansion losses, and

- Reducing mechanical losses by optimizing the mechanical design for the bearing loads involved in the smaller capacities (instead of the larger compressors in the "family" of compressors of which the small compressor is a member).

The OEM cost premium of an improved efficiency, 200 Btu/hr nominal capacity compressor would be on the order of $10 to $20 over the typical current OEM cost of approximately $20 to $25. Regardless of economic payback time, these improvements in small compressor efficiency may be a necessity for meeting the 1993 Federal energy efficiency standard for domestic refrigerators and freezers, or more stringent future standards.

Two generic types of compressors are now in common use in welded hermetic compressors use in domestic refrigerators, reciprocating and rotary. For the smallest required capacities, e.g., for the fresh food compartment of a dual evaporator system, reciprocating compressors are the more promising option for attaining high efficiencies.

9.0 References and Sources

1. Federal Register, April 21, 1989a, pp. 6077-8.

2. Federal Register, November 17, 1989b, pp. 47916-47938.

3. Dieckmann, J. and McMahon, E., State of the Art Survey of Motor Technology Applicable to Hermetic Compressors for Domestic Refrigerators/Freezers. Draft report, prepared by Arthur D. Little, Inc., May, 1990.

4. Tecumseh Products Co., Tecumseh, MI, conference with J. Hackstedd, J. Hammond, T. Kandpal and HU Richardson at Tecumseh.

5. Robert Chilton, Danfoss, Inc., Mahwah, NJ, telcon 11/16/89.

6. Robert Hawley, VP Marketing Americold, White Consolidated Industries, Cullman, AL, telcon 11/22/89.

7. J. Vickerman, Manager GE Motor Marketing, Fort Wayne, IN, telcon 12/7/89.

8. Philip Teague, Chief Hermetic Engineer, GE Small Motor Division, Fort Wayne, IN, conference at Fort Wayne, 1/31/90.

9. ADL report "Study of Energy-Saving Options for Refrigerators and Water Heaters," Vol. 1, May 1977, p. 86, adjusted per inputs from Tecumseh Products Co. (ref 1), and W. Soedel, Herrick Laboratories, Purdue University.

10. M.C.C. Tsao, "Thermodynamics of a Hermetic Reciprocating Refrigeration Compressor with MCCT-Piston and Valve System," ASME Transactions, Vol. 103, April 1981.

11. Hideki Kawai, et al, "The Development of High Efficiency Compressors by Reducing Suction Gas Temperature," Compressor Engr. Division, Matsushita, Ltd., Fujisawa-shi Kanagawa, Japan.

12. GE Small Motor Division, Fort Wayne, IN, conference with H. Harms and D. Erdman at Fort Wayne, IN, 1/31/90.

13. "Impact of a CFC Ban on the Cost and Performance of Household Refrigerators, Centrifugal Chillers, and Commercial/Industrial Refrigeration Systems," Technical Memorandum to U.S. DOE by Arthur D. Little, Inc., August 1980.

14. Nelson, R.T. and MacCarthy, P.W., Research and Development of Energy-Efficient Appliance Motor-Compressors, Final Report, Vol. I Executive Summary and Vol. III Development and Field Test Plan. Columbus Products Corporation (division of White Consolidated Industries), Columbus, OH.

15. Prof. W. Soedel, conversation at Herrick Laboratories, Purdue University, 2/1/90, on compressor efficiencies and limits of rolling piston and scroll compressors.

16. Prof. D. Tree, conversation at Herrick Laboratories, Purdue University, 2/1/90, general developments in the R/F compressor field.

17. Laercio Hardt, Embraco, Calument City, IL office, telcon 5/10/90, compuressor efficiency status.

18. Prof. R. Hill, University of Maine Orono, telcon 4/17/90, LaBreque cycle status, problems with R22 small compressor availability.

19. Constantio Netto, Embraco, Sao Paulo, Brazil, telcon, 3/9/90, status of GE rotaries and problems encountered.

20. J. Sturgeon, GE Compressor Division, Nashville, TN, telcon 3/9/90, availability of improved efficiency compressors.

21. E. Westhoff, White Consolidated Industries (Americold), telcon 3/2/90, advised that Americold does not have compressors with EERs in the 5 range for the lower capacities.

22. J. Poufve, GE Small Motors, Fort Wayne, IN, telcon 1/12/90, obtain data on current motor efficiencies.

23. R. Huffman, Consultant, telcon, 1/4/90, breakdown of R/F manufacturers and their compressor suppliers.

24. T. Nakashima, Sanyo sales office, Chicago, IL, telcon 12/28/89, solicit information on Sanyo compressors, current and future plans.

25. John McCullough, Arthur D. Little, Inc., telcon, 12/28/89, feasibility of scroll compressors in R/F sizes, not optimistic.

26. Mark Okazaki, Matsushita (Panasonic), US rep, telcon 12/12/89, obtain information on Panasonic compressors, current and future plans.

27. J. Hackstedde, Tecumseh Products Co., telcon 11/30/89. Tecumseh makes most motors, buys some from GE.

28. Donald Allen, President Appliance Division, Emerson Motors, telcon 11/30/89, estimates RSIR motors in mid to high 70s, 5-6% more for capacitor run, Emerson no longer supplies to home R/F markets.

29. R. Chilton, Danfoss US Market Manager, tlecon, 11/16/89, supply to GE, Amana, White, Admiral, rotary costly to manufacture. Danfoss not convinced rotary is right direction, pushing 134a as are Matsushita and Sanyo.

30. "Compressors," Ch. 12, 1983 edition ASHRAE Fundamentals.

31. R. Sauber, "Field Testing-Energy Saving Hermetic Compressors in Residential Refrigerators," ASHRAE Transactions, 1982.

32. "The Pressure's On," D. E. Simpson, November 1989, Appliance Magazine.

33. "16th International Congress of Refrigeration," Paris 1983, preprints.

34. T. Shimizu, et al, "Current Status and Trends in Compressor Development," translation from Refrigeration, Vol. 62, No. 720, October 1987.

35. Laercio Hardt, Embraco, presentation at AHAM meeting, Battelle, Columbus, OH, 4/25 & 4/26/89.

36. D. Erdman, et al, "Electronically Commutated Motors for the Appliance Industry," 1984 IEEE publication.

37. "Recent Technological Trends and Lineup of Hitachi Compressors," K. Kikuchi, et al, Hitachi Review, Vol. 36, 1987.

38. Nagatomo, S., et al,"Performance Analysis of Rolling Piston Type Rotary Compressor for Household Refrigerators," 1986 Purdue Compressor Conference Proceedings, pp. 291-298.

39. van der Walt, N., Unger, R., Sunpower Inc., Athens, Ohio, "The Simulation and Design of a High Efficiency, Lubricant Free, Linear Compressor for a Domestic Refrigerator."

40. van der Walt, N., Unger, R., Sunpower Inc., Athens, Ohio, "A High Efficiency, Oil Free, Linear Compressor, Test Results."

41. NASA Refrigeration Meeting.

42. Reefe, R., Americold, telecon 3/23/93, discuss test results of 6.0 EER ECM motor compressor and of Sunpower linear free piston compressor.